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DEFENCE SCIENCE AND TECHNOLOGY ORGANISATION

AERONAUTICAL RESEARCH LABORATORY

MELBOURNE, VICTORIA

Propulsion Report 184

DESIGN AND PRELIMINARY DEVELOPMENT OF AN ENGINE FOR SMALL UNMANNED AIR VEHICLES

by

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DESIGN AND PRELIMINARY DEVELOPMENT OF AN ENGINE FOR SMALL UNMANNED AIR VEHICLES

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SUMMARY

This report describes a program of work aimed at demonstrating the advantages of the opposed piston two stroke engine for use in small, propeller-driven, unmanned air vehicles requiring power of about 10 to 20 kW. Analysis of the requirements for engines for this duty showed the opposed piston configuration as offering the best compromise in terms of specific output, specific fuel consumption, level of vibration and complexity.

Initial feasibility was demonstrated using a small engine of 67mL swept volume. The major part of the development, including selection and development of a scavenge air compressor, was carried out using an engine of 225 mL swept volume. The maximum power obtained was 18.9 kW (25 HP) at 7500 rpm and the engine is considered to have potential for considerable further development.

Considerations involved in the selection of the configuration are discussed. The development of the rig engine and of the scavenge air blower is described and details of the performance are given. Factors limiting the performance and the potential for increased output and improved economy are outlined.

During the course of this work the initial feasibility of a more radical design, with the connecting rods and crankshafts replaced by a system of cams and rollers, was also demonstrated by making and running an engine. This arrangement offers the prospect of saving about 15% in size and weight compared to the conventional engine of the same swept volume.

Work on the cam engine is described in an appendix to the main report.

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1. INTRODUCTION

The earliest unmanned air vehicles (UAVs) were conventional aircraft, modified to permit operation of the controls via a radio command link. As the military potential of UAVs became more apparent, changes in payload and range requirements resulted in the evolution of a class of vehicles of a much smaller size, usually powered by a reciprocating engine driving a propeller.

During the 1960s, selection of engines for this type of UAV was done simply on the basis of commercial availability. As a result, engines designed originally for use in chain saws, portable generators, lawn mowers etc. were used and quickly found to be unsatisfactory in terms of life at high output, level of vibration, fuel economy and reliability.

Subsequently, the first generation of engines designed specifically for the purpose appeared. In order to minimise both cost and technical risk these designs were usually based on mass produced components (sometimes from a number of different manufacturers) assembled so as to produce a configuration better suited to the purpose, most commonly an arrangement of two opposed cylinders, which were offset to accommodate the two adjacent crank throws.

More recent military applications for UAVs, such as laser target designation, real time television surveillance (using long focal length lenses) and infra-red imaging, necessitate the use of engines having very low levels of vibration in order to meet stringent requirements for aiming accuracy and image resolution. This, together with the ever-present requirements for high specific output and good fuel economy, warrants a reappraisal of the choice of engine configuration.

Selection of the most appropriate configuration of piston engine for the duty is very important. Since all engine design involves compromise, failure to optimise the engine for a specific duty will involve penalties in terms of vehicle weight, range and endurance. There are some indications that, in the foreseeable future, there are likely to be sufficiently large numbers of UAVs requiring engines of this size to justify the ab initio design of an engine for the purpose.

The following sections describe the preliminary development of such an engine in the 20 kW class, which could propel an aircraft with a wing-span of about 2-3 metres, flying at 40 to 60 m/s.

2. SPECIAL REQUIREMENTS FOR UAV ENGINES

It is not possible to rank all the required engine characteristics in order of importance without knowing the vehicle characteristics, mission profile, method of launching, whether or not the vehicle will be recovered, the type of fuel to be used and the numbers required. However it will be useful to discuss, in general terms, the (usually conflicting) requirements which are common to most applications and which have a strong effect on the choice of configuration.

These are:

High power/weight ratio

Low specific fuel consumption

Low level of vibration

Simplicity, low cost

Power output in the range 15-25 kW

Ability to use the specified fuel

Other characteristics which are rather more dependent on detailed design, but which are nonetheless influenced by the configuration are:-

Reliability (including starting)

Low acoustic and infra-red signatures

Torque/speed characteristics to suit propeller

Many of these characteristics are of course common to all engine applications. However, the ranking of these and the weighting to be given to each results in a list peculiar to the UAV application. This is the reason for the initiation of the programs of engine development which began in both the UK and USA in the late 1970s and early 1980s, since no commercially available engines had the precise combination of attributes required. (Reference 1)

The range of power outputs listed above effectively rules out the use of gas turbine engines, since the efficiency of gas turbines of this size is extremely low. Nor have rotary combustion engines of the Wankel type yet been able to demonstrate sufficiently good specific fuel consumption to compete with piston engines in this size range.

The ability of an engine to burn a specified fuel is obviously an important consideration. In a recent request for proposals for engines for a propeller-driven RPV, the US Navy has specified the use of JP 5 or diesel fuel for logistic reasons, despite the anticipated performance penalties compared with gasoline engines.

In this context, the recent development by the Orbital Engine Company of direct in-cylinder fuel injection for two-stroke engines is clearly of great significance, since they have been able to demonstrate a very considerable reduction in specific fuel consumption. Their air-assisted atomising system would be expected to lend itself to adaptation to heavier fuels, although lower compression ratios may be required in order to accommodate the lower knock rating of these fuels, with a consequent reduction in performance.

It follows therefore, that in order to meet anticipated logistic requirements and to be competitive in terms of specific fuel consumption, any future, purpose-designed two-stroke UAV engine should preferably be

equipped with direct in-cylinder injection and the configuration should be selected with this in mind.

3. SELECTION OF CONFIGURATION

3.1 Selection of Engine Type

The maximum flight speed for mini-RPVs is typically of the order of 50-80 m/s, at which speed either a propeller or ducted fan would be selected for propulsion on the basis of propulsive efficiency. The size of the propeller or fan will usually be limited by the constraints of handling or storage as well as the requirement to clear the ground or launcher, so that relatively high speeds of rotation will be required. This consideration, together with the requirements for high power/weight and simple and cheap construction, lead to the selection of the two-stroke cycle, while weight considerations dictate the use of an air-cooled design..

3.2 Arrangement and number of cylinders

The next decision to be made concerns the number and disposition of the cylinders. When using gasoline fuel, the requirement for small cylinders compared with those used in general aviation is to some extent beneficial, since the increased surface to volume ratio of the combustion space reduces the tendency to detonation. However, in the small sized engines involved here, the swept volume per cylinder, and hence the size of the combustion chamber, becomes less than is desirable as charge chilling becomes a significant consideration, especially when multi-cylinder arrangements are favoured for good balance of inertia forces.

A quantitative appreciation of the effect of cylinder size on specific fuel consumption (in diesel engines) may be gained from Figure 1, reproduced from Reference 2. In the present context, even a single cylinder engine would have a bore much less than 100 mm in diameter so that, if the number of cylinders is increased to two, thus reducing the diameter to 70% of its original value (other things being equal), the specific fuel consumption will be greatly increased. In the case of the diesel engine this effect is partly due to charge chilling and partly due to impingement of the fuel spray on the cylinder walls. The latter effect will also be of significance in the case of a direct-injected spark ignition engine.

3.3 Selected Configuration

The configuration proposed, and on which the development program was based, is the single cylinder, opposed piston, two-stroke. This configuration is rarely used for small engines, probably because it does not lend itself to the use of crank case compression for the scavenge air and therefore necessitates the use of a scavenge air compressor. It does, however, have the attribute of combining the relatively larger combustion chamber associated with a single cylinder with the balancing of inertia forces associated with an opposed cylinder twin (without the out-of-balance couple arising from the offset cylinders).

The principal features of this configuration are shown diagrammatically in Figure 2. It comprises a common cylinder in which two pistons with dished crowns approach one another to form a single, lenticular, combustion space. At opposite ends of the cylinder are two crankshafts, which are interconnected (in this case by a toothed belt). The charge air enters one end of the cylinder and the burnt products are exhausted at the other.

3.4 Advantages of the Configuration

This scavenging arrangement is known as "through flow" or "uniflow" scavenging and is usually acknowledged as the most desirable. It reduces "short circuiting", that is, the escape of the fresh charge air through the exhaust before the burnt products have been purged from the cylinder. It also helps to reduce the intermixing of the incoming charge with the combustion products, so giving a higher charge density, and hence increased output.

In theory, this configuration offers the prospect of perfect balance of all forces and couples, provided the cranks rotate in opposite directions and in phase. In this case the vertical shaking force (considering the cylinder to be horizontal) due to the motion of the connecting rods can be cancelled by weights rotating with the two crankshafts. This would of course require the crankshafts to be connected by gears. This is not an attractive choice in the present application because of the cost and weight involved and because of the difficulty of air cooling that part of the cylinder shrouded by the gears.

The use of a toothed belt results in a lighter and cheaper arrangement in which the cranks turn in the same direction so that the connecting rods cause a small out-of-balance couple. This is regarded as the better option. In the large opposed-piston engines in service at present, it is usual to set the exhaust crank with a phase advance of about 15 degrees relative to the inlet crank so as to achieve asymmetric exhaust port timing. The significance of this is that the exhaust port which, as is usual, opens before the inlet port, is made to close before the inlet and so give an increased charge trapping efficiency at the expense of some out of balance of shaking forces acting along the cylinder centreline. This is fairly typical of the compromises which must be made in engine design and serves to illustrate the point that some final design decision can only be made in the light of particular requirements.

In the case of a direct-injected engine, (air-) charge trapping efficiency is not quite so important, because the cylinder scavenging is effected by air only, and not by a fuel/air mixture, as is the case with a carburetted engine. If, for a particular application, the absolute minimum level of vibration were required, it is probable that symmetrical timing could be used without a great performance penalty.

As remarked above, the single combustion chamber has a lower surface/volume ratio than the two combustion chambers of an opposed cylinder arrangement. The single combustion chamber is also an advantage if direct injection is to be used, because of the lower weight and cost associated with only one injector and also because of the reduced chance of fuel impingement on the walls.

Location of the ports at opposite ends of the cylinder makes the whole of the circumference available for both the inlet and exhaust ports, unlike cross or loop scavenged engines. This has the effect of reducing the port height, so making a greater part of the stroke available for useful work. It also facilitates the use of inlet swirl to control the air movement in the cylinder, although this was not used in the present engine.

The disadvantages of this configuration are the necessity for a scavenge compressor, the extra crankshaft and bearings, and an increased heat load on the pistons, particularly the exhaust piston, because they are not exposed to the cooling effect of the scavenge air on their undersides, as is the case with crankcase scavenged engines.

On the other hand, the use of a scavenge compressor means that positive crankcase lubrication may be used to cool the piston and this permits the use of plain bearings for the crankshaft and connecting rod ends. Plain bearings are both cheaper and lighter than rolling element types.

In cases where the propeller or fan is to be directly driven the asymmetry of the engine may be inconvenient, although it is well suited to pylon mounting of the propeller as shown in Figure 3. Where indirect drive is to be used, either to achieve speed reduction, or because of a requirement to bury the engine in the fuselage in order to minimise IR or radar signatures, the arrangement shown in Figure 4 may be used. This allows the engine to be mounted on the vehicle centreline and gives a compact drive, with the power transmission shared by two belts.

4 SCAVENGE AIR COMPRESSOR

Inspection of Figure 2 shows the difficulty of using the conventional crankcase compression in order to provide the scavenge air. The inlet crankcase alone would not provide sufficient air and the exhaust crankcase would have to be used as well. The necessary passages would obstruct the cooling airflow to the cylinder and complicate the assembly and manufacture. Evidently some other provision must be made to supply the scavenge air.

4.1 Requirements for Scavenge Air Compressor

The basic requirements for the compressor are that its output should be matched to the engine demand over the range of operating speeds, it should be efficient, small in size and weight, cheap to make and, in order to avoid the use of extra gears or bearings, should be driven at engine speed.

Selection and subsequent development of an appropriate scavenge pump for use on the engine absorbed a significant part of the total development effort. This was due in part to the fact that, as the basic engine was undergoing development, and port heights and the crank phase relationship were changed, the flow characteristics also changed, so changing the flow from the compressor, and hence the scavenging ratio.

The scavenging ratio is defined as the mass of air supplied to the cylinder per stroke divided by the mass required to just fill the cylinder with fresh air at inlet manifold temperature and exhaust pressure. In general, greater scavenge ratios lead to greater scavenging efficiency, that is, the charge retained in the cylinder becomes less diluted by combustion products. References 3 and 4 discuss this matter more fully. Crankcase scavenged engines rarely achieve a scavenge ratio of unity and considerable effort is usually made to maximise it by the use of unsteady gas dynamic effects and disc or reed valves at inlet to the crankcase.

In the case of a positively scavenged, carburetted engine, the value of the scavenge ratio to be used is more at the discretion of the designer and a compromise must be sought between thorough purging of the cylinder by the use of an excess of air on the one hand, and the power consumed by the compressor and loss of unburnt fuel through the exhaust port, on the other.

The use of a carburettor calls for the use of the lowest possible scavenge air ratio consistent with good scavenging, because the cylinder is purged with a mixture of fuel and air whereas, when fuel injection is used, only air is involved. For the demonstration engine a scavenge ratio of 1.2 was the target figure.

For a given compressor flow rate, the power requirements can be minimised by keeping the delivery pressure as low as possible. The pressure is determined by the area of the ports, their discharge coefficients, the proportion of time for which they are open, and by unsteady gas dynamic effects in the exhaust and inlet passages.

There is of course a large body of literature on the subject of two-stroke scavenging and the use of unsteady gas dynamic effects to improve it. However most of the recent literature, and particularly the computer models of the process, apply to the crankcase scavenged engine.

4.2 Types of Scavenge Air Compressor

The scavenge air requirements of large marine and industrial uniflow scavenged engines are usually met by shaft-driven compressors of centrifugal or vaned rotor design, or lobed types such as the Lysholm or Roots blowers. The centrifugal types require the use of a speed increasing gearbox but are otherwise simple and compact. The flow characteristics of centrifugal compressors are such that the air supplied per revolution of the engine, and hence scavenging ratio, varies with engine speed. The vaned or lobed types however, which have near-positive displacement characteristics, enable the scavenging ratio to be held nearly constant over a range of speeds.

The lobed compressors vary in detailed design but share the common feature that they have two parallel rotors with intermeshing sets of lobes, either straight or helical. In general, compressors of this type must be made very accurately, including the gears which maintain the angular relationship between the rotors. The resulting high cost, together with their heaviness, rules them out of contention for the present application.

4.3 Selected Scavenge Air Compressor

During the course of the preliminary experiments, for which a 67 mL engine was constructed, a centrifugal compressor of 50 mm diameter, with a design speed of 65,000 rpm, was built. This was driven from the crankshaft at the inlet end of the engine by means of an epicyclic friction drive which also supported the rotor. Great difficulty was experienced in starting the engine with this arrangement, since the phase advance of the exhaust crank resulted in the engine having a slight reverse pumping action which was enough to overcome the very small pressure rise of the centrifugal compressor at cranking speed. The centrifugal impeller is shown in Figure 5, with the volute omitted for clarity.

Subsequently, a novel type of vaned compressor was used to achieve more promising results with the small engine and then was used extensively with the larger engines. In fact, most of the development of the larger engine was carried out using compressors of this type, which were progressively modified in the course of the work. This will be referred to as a trailing vane compressor.

The operating principle may be seen by reference to Figure 6 which shows an early version of this type, with the inlet and outlet ports located in the cover plate. A cylindrical rotor is eccentrically mounted in a cylindrical housing such that its outer surface is close to the inner wall of the housing at one point. Three, or more commonly, four vanes are pivoted at the outside surface of the rotor in such a way as to be flung outwards by centrifugal force as the rotor turns, and to lie flat against the rotor as they pass that point where the clearance between the rotor and the housing is a minimum. The direction of rotation is such that the pivot of the vane leads and the outer end of the vane, which is in contact with the housing, trails. The space defined by the trailing face of one vane and the leading face of the next, and by the housing and rotor surfaces, at first increases and then decreases as the vane passes from the inlet to the outlet ports, located on either side of the point of minimum clearance.

With this kind of compressor some internal compression takes place, unlike say, the Roots blower, which is a constant volume displacer. The extent of this compression can be chosen by appropriate location of the outlet ports. It is of course important to avoid over-compressing the charge, that is, above the manifold pressure, since the resulting expansion will be unopposed and energy lost.

This type of compressor will deliver, in one revolution, a greater volume than that contained between the rotor and the housing, although theoretical prediction of the volume delivered is made very difficult by the unsteady gas dynamic effects. However, it has been found in practice that, when supplying the engine, the volume of air delivered per revolution at inlet conditions is of the order of 1.2 to 1.3 times the volume of the space between the rotor and housing.

The principal advantage claimed for the trailing vane configuration was that the pressure difference acting across the vanes opposed the centrifugal force and so tended to lift them from the wall of the housing. It was anticipated that this would have the effect of reducing wear and power loss due to friction and would result in efficient operation. The effect of the pressure difference on the vanes could be demonstrated by blocking the output while the compressor was

being driven, until the vanes lifted from the walls and the flow was reduced to zero. Upon removal of the blockage the flow resumed without any ill effects.

Despite fairly extensive development however, it became apparent that the hoped-for efficiency would not be achieved. The poor efficiency resulted in the delivery of air at high temperature to the engine, giving reduced output and leading to detonation and mechanical distress in the engine generally. It was also found to be very difficult to achieve a satisfactory life for the pivots, which were highly loaded. Subsequent analysis of the motion of the vanes and forces acting on them by Brizuela, and later by Parmington, showed that the anticipated reduction of the rubbing loads actually occurred over only a small fraction of the cycle and so had only a small effect of the power requirement. (Refs. 5 and 6)

In order to achieve a higher compressor efficiency and hence lower engine inlet temperatures it was decided to investigate the use of sliding vane compressors, of which there are several different types. They all involve vanes which slide in slots in a rotor which is eccentrically mounted in a housing. Some designs incorporate mechanical arrangements to prevent contact between the vanes and the wall of the housing so as to minimise rubbing friction, although this was considered to be an undesirable complication in the present application.

Designs in which the vanes contact the wall may have a number of discrete radial vanes or they may have one or two diametral vanes which pass through the centre of the rotor (where they also pass through one another). A theoretical analysis, carried out by Mabanta (Reference 7) as part of the engine development program, showed the friction power taken by a single diametral vane to be less than that taken by a pair of discrete radial vanes.

The design finally selected is shown (with the end cover removed) in Figure 7. It utilised a single diametral vane in a circular housing. The decision to use a single vane was made on the basis that, although the volumetric efficiency would be slightly less than would be the case for two vanes, the friction losses would be smaller. The use of two vanes would double the friction load but could not be expected to give a corresponding increase in the volume flow rate. It was recognised that the use of the circular housing rather than a cardioid shape would result in higher peak accelerations of the vanes, but this was accepted in order to simplify manufacture.

5. ENGINE DETAILS

5.1 67 mL Engine

A preliminary assessment of the feasibility of the opposed piston configuration was carried out using an engine of 67 mL swept volume. Two such engines were made using cylinders machined from bar stock aluminium and having hard anodised bores. Figure 8 shows one of the engines mounted on the test bed.

The decision to use hard anodising for the bore was made on the basis of advice that its life would be adequate for the purpose and that it would be

cheaper and have better heat transfer characteristics than a sleeved bore. However, it proved to be totally unsatisfactory as the surface began to disintegrate within the first hour of running. Notwithstanding this, the initial feasibility was established and the port dimensions and the exhaust crank phase advance shown to be such as to at least enable the engine to run. Most of the running with this engine was done with the scavenge air supplied by a separate, electrically driven centrifugal blower.

5.2 225 mL Engine

The great majority of the test running was carried out using two (initially identical) engines of 225 mL swept volume. The bare engine, without inlet manifold, scavenge compressor or exhaust pipes is shown in Figure 9. Figure 10 shows the engine mounted on the test bed, coupled to an eddy current dynamometer.

The design philosophy was that, since it was only the basic attributes of the opposed piston configuration which were to be demonstrated, certain compromises in detailed design would be adopted in the development version in order to withstand the rigours of test bed development, such as running for extended periods at wide open throttle. No attempt would therefore made to produce an engine which could be installed in a flight vehicle.

Although the configuration of the engine was chosen so as to readily accommodate the use of direct in-cylinder injection, it was decided to proceed initially with a carburettor in order to keep the engine as simple as possible during the initial development phase. The intention was that installation of fuel injection should form part of a later phase of development, after the validity of the configuration had been demonstrated.

Based on the experience with the smaller engines, it was decided to use a centrifugally cast iron cylinder liner shrink-fitted to a wrought aluminium cylinder which was machined from bar stock. This construction was selected for its general robustness and resistance to damage in the event of piston seizures, but would of course be unacceptably heavy and expensive for a flight version. Furthermore, the interface between the liner and the surrounding finned cylinder is known to impede heat flow. For flight engines in batch production, the use of cast cylinders using a material such as the Comalco casting alloy 3HA, would now be the preferred method, since the manufacturers assert that pistons can be run directly on an appropriately machined bore without further treatment.

The bore was 56 mm, the stroke 45.72 mm and the connecting rod length was 100 mm. There were six inlet ports uniformly spaced around the cylinder. Each port was 19.7 mm wide and, as originally machined, was 9.1 mm high. This was later modified to 10.5 mm high on one engine. The inlet ports occupied about 67% of the cylinder circumference.

There were four exhaust ports, arranged in two pairs to facilitate connection of the exhaust pipes. Each port was originally 22.5 mm wide and 10.5 high. This was later modified to give one engine with both inlet and exhaust ports 10.5 mm high, and the other with inlet and exhaust ports 9 mm and 13 mm

high respectively. The exhaust ports occupied about 61% of the cylinder circumference.

The pistons and crankcases were sand cast in Aluminium alloy LM 14 to BSS 1490. They were originally fitted with two rings but towards the end of the program an oil control ring was fitted near the bottom of the skirt in order to prevent loss of oil by leakage from the inlet crankcase past the piston skirt into the inlet manifold.

The piston rings, piston pins, connecting rods and connecting rod needle-roller bearings were mass produced components for the Honda 250 series motorcycles. Lateral location of the connecting rods was by the crank webs, using silver plated thrust washers.

The crankshafts were of built-up construction, using a specially designed tapered draw-bolt through the crank pin to produce a tight fit in the crank webs. The material used for the crankshafts was EN 36A alloy steel, ground after heat treatment to 60 Rockwell C hardness.

The crankshafts, which were carried in needle-roller bearings, were coupled by means of a toothed belt, Uniroyal HTD series, having teeth at 8 mm pitch. The belt width was 50 mm. No provision was made for adjustment of belt tension. The accuracy of manufacture of the belts was such that this was not required. The belt pulleys each had 34 teeth and were keyed to the crankshafts in such a way that the phase relationship between the crankshafts could be changed in increments of one half of one tooth pitch, that is, 5.6 degrees. The pulleys were originally made from aluminium, but for extended rig testing this was replaced by steel.

With the opposed piston configuration, changes in the phase relationship between the cranks result in changes in the compression ratio and, to a small extent, in the displacement volume. For this reason care had to be taken to select the correct compression height (the height of the piston crown above the gudgeon pin centreline) for the pistons for each value of phase advance, so as to prevent detonation. The relationship between exhaust crank lead and both compression ratio and displacement volume is shown in Figure 11. The compression ratio shown is that calculated by taking the maximum volume (which, in an opposed piston configuration employing phase advance, occurs when the pistons are symmetrically located about bottom centre) in accordance with British and American practice. European practice is to calculate the maximum volume at the instant the ports close.

The engine was lubricated by an electrically driven oil pump which delivered oil to jets aimed at the underside of the piston crowns. The crankcase drained by gravity back to the external sump. In the early phases of the development lubricating oil was also added to the fuel as a precautionary measure, although this is known to reduce the effective knock rating of the fuel and its use was discontinued towards the end of the project. A very small amount of oil was supplied to the scavenge compressor through the centre of the shaft and this found its way into the cylinder.

No attempt was made to integrate the scavenge compressor with the engine as would be the case with a flight worthy design. This had the advantage that the compressor could be modified or replaced without dismantling the engine. The mixture was fed from the compressor to the annular inlet manifold through welded steel tubing which incorporated a flexible joint.

Carburation was by means of a straight-through carburettor using a throttle slide and needle for fuel metering. Mikuni and Keihin motorcycle carburettors were used at various times, having throat sizes of 20 to 23 mm diameter. Maximum power output was obtained with the 23 mm throat and a main jet diameter of 1.62 mm.

Two spark plugs on opposite sides of the cylinder, at the centre of the engine, provided the ignition. High plug temperatures resulting from the engine design and the use of fairly high compression ratios required the use of "cold" plugs. Prior to the installation of the oil control ring at the bottom of the piston skirt it was necessary to change plugs after the engine was warmed up. Champion J4J plugs were used for starting from cold, after which surface discharge plugs (L20V) were used for wide-open-throttle running. The J4J plugs became hot enough to cause preignition under high heat loads but were necessary to prevent fouling under starting conditions. Following the installation of the oil rings, the surface discharge plugs could be used throughout the test runs.

Surface discharge plugs are usually recommended for use only with capacitor discharge ignition systems. However, due to the very fast rise times involved, electromagnetic radiation from the capacitor discharge ignition was found to interfere with the electronic data acquisition system and it was replaced by a conventional transistor-switched coil ignition, using two automotive coils.

Spark timing, which could be manually adjusted during running, was from a disc attached to the connecting shaft between the engine and the dynamometer. A slotted opto switch, comprising a Ga As infra red LED coupled with an npn silicon photo-transistor, was activated by the gap in the timing disc and fired both plugs simultaneously.

In the case of an opposed piston engine with exhaust crank lead, spark advance is affected by changes in the lead angle. This is because minimum volume (to which the spark advance must be referred) does not occur when either piston is at top centre, but when the inlet and exhaust cranks are symmetrically located either side of top centre by an angle equal to half the exhaust crank lead angle.

6 TESTING ARRANGEMENTS

6.1 67 mL Engine

Limited performance testing of the 67 mL engine, shown in Figure 8, was carried out using a specially built test bed incorporating a Go Power water brake

dynamometer, originally intended for use with Go Kart engines. The engine was lubricated by hand. The only measurements taken were speed, torque and inlet and exhaust temperatures.

6.2 225 mL Engine

Most of the performance measurement with the larger engine was carried out using an eddy-current dynamometer (Ozy Dyn Model EC 101), although some early results were obtained using a Heenan and Froude water brake dynamometer model (DPX0). Figure 11 shows the test set-up. The engine was connected to the dynamometer by means of a torsionally resilient quill shaft. This was found to be necessary because, with a stiff shaft, the relatively large inertia of the eddy current dynamometer rotor caused high transient loads which resulted in a series of failures of the crankshaft and connecting shafts.

Clearly, the type of exhaust system fitted to the engine will have a considerable bearing on the performance. However, since the objective was to demonstrate other aspects of the configuration, as few changes as possible were made to the exhaust system. Much of the test running associated with the development of the scavenge air blower was carried out using the standard exhaust pipes from the Honda 250 motor cycle engine. These were of the non-resonant acoustic absorption type (Burgess Tube). During the later tests, after the sliding vane scavenge air compressor was adopted, a simple tuned exhaust system referred to in-house as the Mark 4, was fitted. As will be seen this gave marginally better performance than the Honda type. However, no attempt was made to optimise it for the particular combination of exhaust crank lead and port height in use at the time.

Cooling air flow to the engine was supplied by an electrically driven centrifugal blower through flexible ducting, visible in the photograph.

Engine speed was measured by means of a toothed wheel attached to the output shaft. The retarding torque applied by the dynamometer power absorption unit was measured by a strain gauge load cell which was connected via a torque arm to the trunnion mounted stator. Calibration was by means of known weights hung on the torque arm directly above the load cell.

The dynamometer was equipped to permit closed loop control of speed or torque. All the performance testing was done in the constant speed mode, so that torque was automatically varied to keep speed constant at the selected value. Water flow rate to the dynamometer was monitored so that, in the event of failure of the supply, the ignition was disabled.

Fuel for the engine was standard grade automotive gasoline containing Lead Tetraethyl. The fuel flow was measured by means of a positive displacement Fluidyne flow meter, providing both flow rate and total flow outputs. Calibration was by comparison with a laboratory burette.

Air flow rate to the engine was measured by means of a positive displacement rotary gas meter, chosen for its insensitivity to pressure pulses in the airflow. The intermeshing rotors of this meter had significant inertia and care was taken to establish steady running conditions before airflow

measurements were recorded. Use of the constant speed facility incorporated in the dynamometer made this a relatively simple matter.

The temperature of the air entering the engine and entering the air flow meter, as well as other temperatures recorded from time to time, were measured by Chromel Alumel (Type K) thermocouples connected via Analog Devices thermocouple transmitters to the data acquisition system.

The data acquisition system comprised a Computer Plus LSI 11-73 computer running a DAOS 6.2 data acquisition software package.

A standardised procedure was adopted when recording a set of performance data. With the dynamometer operating in constant speed mode and the throttle wide open, the required speed was set on the dynamometer control and the spark timing was then adjusted manually in order to give the highest torque figure. When this had been obtained and the engine speed was steady, a period of about 15 seconds was allowed for the airflow meter to stabilise and then the set of data was recorded.

Each of the performance parameters was digitally sampled 128 times at 1 msec intervals and then averaged to generate one test point.

7. PERFORMANCE OF 225 mL ENGINE

The engine performance figures have been corrected to Standard Atmosphere equivalent value, according to A.S 1501 2.3.3 (Rating and Testing Internal Combustion Engines), that is, to an inlet air temperature of 300 K and ambient air pressure of 100 kPa.

Maximum power output was 18.9 kW (25 HP) at 7400 RPM and the peak torque 24 Nm, corresponding to a brake mean effective pressure of 670 kPa (97 p.s.i). Figure 12 shows the variation of torque and power with speed. For this run the engine was equipped with 9 mm inlet ports and 13 mm outlet ports, the sliding vane scavenge air compressor, the Mark 4 exhaust system and set up with 21 degrees exhaust crank lead and optimum spark advance.

Figure 13 shows, for the same conditions, the variation in equivalence ratio and specific fuel consumption. Examination of this figure, in particular the discontinuity between 5500 and 6000 RPM, shows the direct dependence of specific fuel consumption on the equivalence ratio provided by the carburettor. A persistent malfunction of the carburettor which is evident in this speed range is also believed to be the explanation for the corresponding discontinuity in this part of the torque curves generally, although attempts to eliminate it by changing the carburettor, the length of inlet pipe, and the volume of the inlet manifold, were unsuccessful.

The best result obtained with the trailing vane compressor is compared with the best result for the sliding vane compressor in Figure 14, which shows, in terms of torque and power output, that the sliding vane compressor gave

marginally better performance above about 7000 rpm. The beneficial effect of the improved compressor is masked in this comparison by the fact that this compressor, the only one of its type which could be built during the life of the project, was slightly too small. This may be seen in Figure 15 (despite an unexplained excursion in the curve for the trailing vane compressor), in which the scavenging ratios in the same two test runs are compared. As noted above, the target value for scavenging ratio used in sizing the compressors was 1.2. However, the difficulty in estimating the volumetric efficiency to be expected from the compressors when installed on the engine resulted in the trailing vane type being marginally too big, and the sliding vane type too small.

In separate rig tests, in which the compressors where driven by an electric motor via a transmission dynamometer, the sliding vane type was found to be much more efficient. On the engine, improved compressor efficiency is reflected in both lower power consumption to drive the compressor and in the lower delivery temperature, which increases the charge density in the cylinder at the start of the compression stroke and so gives greater power output. The sliding vane type produced about the same engine output from a smaller airflow rate because it is more efficient. The lower temperature is also clearly desirable in order to avoid excessive engine temperatures at high output.

The effect of the compressor performance on the engine specific fuel consumption for the same set of data is shown in Figure 16. Although once again the rich mixture provided by the carburettor at about 6000 rpm tends to mask it, it is apparent that there is a very significant effect, particularly at the highest speeds.

The sliding vane compressor is clearly to be preferred. If it were to be made slightly bigger, it is reasonable to assume that its adiabatic efficiency would remain about the same, in which case engine output would be increased without detriment to specific fuel consumption.

The effect of changes in exhaust crank lead on output, for the case of 10.5 mm inlet and 10.5 mm outlet ports, is shown in Figure 17, and for the case of 9 mm inlet and 13 mm outlet ports, in Figure 18. In both cases the same (sliding vane) blower, the same carburettor and the same exhaust system were used. Because changes in exhaust lead also changed the displacement volume and, much more significantly, the compression ratio, the range of phase angles which could be investigated was limited by the availability of pistons having appropriate compression heights so as to limit the compression ratio and avoid the onset of detonation.

Figure 19 shows, other things being equal, the effect of the exhaust configuration on performance. The Honda exhaust was of the non-resonant type. The so-called Mark 4 was a resonant type, designed for peak performance at 7000 rpm, with the objective of achieving a rising torque curve in this region. The reason for its marginally better performance over the rest of the speed range is not known.

8. DISCUSSION

A prominent feature of the results is the generally flat, or slightly rising torque curve which is maintained to the maximum speed and which results in a steadily rising power versus speed relationship. This is regarded as a clear vindication of the positively scavenged, opposed piston configuration. The maximum power output is currently limited by the strength of the engine resulting from the available manufacturing techniques and significant increases could be expected with improved techniques and materials.

The major influence on the overall engine performance is the performance of the scavenge compressor, a factor which was recognised at the outset as being critical to the success of the concept. The relatively small changes in output which result from changes in port heights and exhaust system seem at first sight to be surprising in the light of the profound effect which such changes are known to have on the performance of cross or loop scavenged engines. However, a possible explanation, which would also apply to the relative insensitivity to changes to exhaust crank lead, is that, in the case of a positively scavenged engine, the airflow rate, and hence scavenge ratio, are more or less completely determined by the scavenge compressor. Thus the unsteady gas dynamic effects due to the exhaust system and the timing of port openings, act through changes in the charge trapping efficiency to produce less pronounced changes in output.

The results show only the wide open throttle performance, since the carburettors used could not be adjusted to give an even approximately constant fuel/air ratio over a range of throttle openings. Any attempt to produce a performance map therefore gave results which reflected carburettor performance rather than engine performance. This effect is apparent even in the wide open throttle performance as can be from the results. The main jet size and the needle position were selected to give maximum power at wide open throttle, without regard to specific fuel consumption.

The performance limitations resulting from the simple construction of the carburettors were accepted for the purpose of demonstrating the feasibility of the configuration. However, the intention was that, in the next stage of development the engine would be fitted with direct, in-cylinder, fuel injection.

9. SUGGESTIONS FOR FURTHER DEVELOPMENT

Clearly, there was scope for further development of the engine in order to improve performance and mechanical reliability. However, since the anticipated specific application failed to materialise, further development of the engine will only be undertaken with some new application in mind and the direction of the work will be largely determined by the constraints arising from this application. However, there are some observations, arising from the development so far, which will be of value.

Probably the highest priority in further development would be to optimise the matching of the scavenge compressor to the engine airflow demands. The air flow delivered must be adequate for the cylinder to be scavenged and yet not so great as to consume excess shaft power. The target figure for scavenge air ratio used with the carburetted engine in the present program must be regarded as arbitrary and would need to be varied according to whether or not direct injection was to be used.

At the stage of development when the engine was producing the greatest power and cylinder pressures were therefore correspondingly high, a series of piston failures were encountered involving collapse of the crown. Metallurgical examination showed that the castings were porous, particularly in this critical region, and that this was the cause of failure, rather than overheating or detonation. It was considered unlikely that any significant improvement in quality could be achieved without abandoning the sand casting method of production and using die-cast or forged pistons.

Sand cast pistons were used in the rig engine because pistons with a dished crown were not available as mass produced items. The dished crown was thought to be desirable so as to form a lenticular combustion space and to produce a squish effect as the pistons closely approached each other. However, the importance of the dished crown was not experimentally demonstrated and it may be possible to use flat topped, mass-produced pistons without a significant performance penalty.

As remarked earlier, an engine designed for production and subsequent flight use would incorporate the blower housing and inlet manifold as an integral part of a single, light alloy, cylinder casting. This would reduce the parts count and, if the pistons could be run directly on the bore without the use of a liner, would also improve the cylinder cooling, partly because of the improved thermal conductivity compared with the cast iron used for the liner, and partly by eliminating the interface between the liner and the outer cylinder.

Following recent demonstrations of direct injection two stroke engines for automotive use, it seems likely that the next generation of high performance military two stroke engines will be equipped with direct injection in order to achieve good economy, low emissions and possibly a multi-fuel capability. In addition to the system developed by the Orbital Engine Company in Australia, there are said to be competing systems under development in West Germany, UK and USA.

Given that further development of the engine would preferably involve the use of direct in-cylinder fuel injection, it may be possible, at the expense of some loss of performance, to eliminate the phase advance of the exhaust piston and so achieve improved balance of the inertia forces. Naturally this would have a considerable effect on the scavenging process and the port heights would have to be changed. The use of direct injection would mean that fuel would not be lost with excess scavenge air; however, care would still be required to match the scavenge compressor output to the engine requirements so as to minimise pumping losses.

10 CONCLUSION

The validity of the proposed configuration has been adequately demonstrated, the basic design features and the optimum type of scavenge air compressor have been identified, and a clear indication of the direction for further development provided. In the opinion of the authors, this configuration, fitted with in-cylinder fuel injection, must be considered an attractive choice for a UAV requiring power in the 20 kW range.

11. ACKNOWLEDGEMENTS

The authors wish to acknowledge the skilful and patient manufacturing and assembly work carried out by Mr David Dyett throughout this project. In particular his cheerful acceptance of the inevitable seizures and other failures during performance test runs, and his constructive suggestions for design improvements, have been most valuable.

Thanks are also due to Mr Peter Gage for initially setting up the data acquisition software and to Mr Arthur Mabanta for continued support and development in this area.

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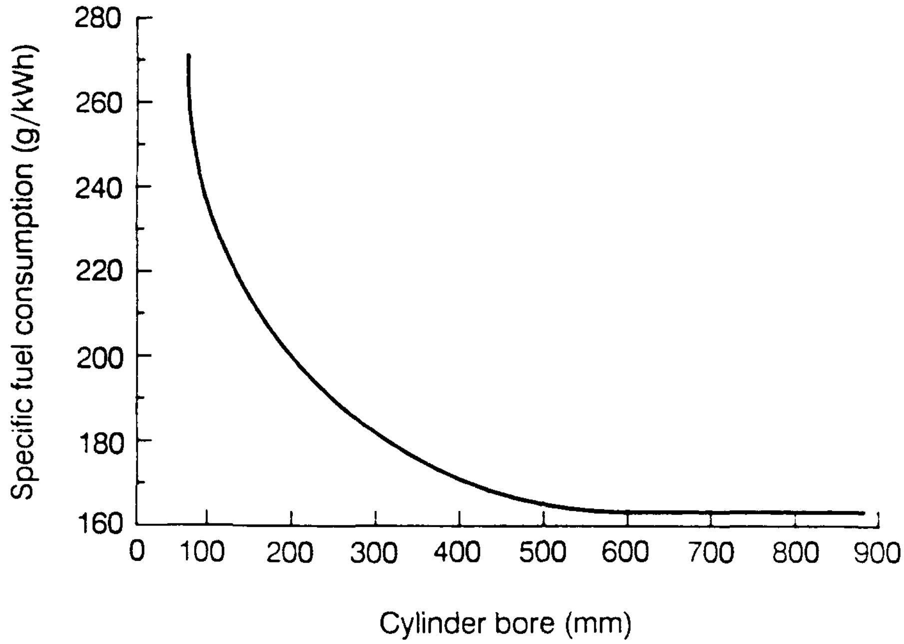


FIGURE 1. EFFECT OF CYLINDER SIZE ON SPECIFIC FUEL CONSUMPTION (FOR DIESEL ENGINES) (FROM REFERENCE 7)

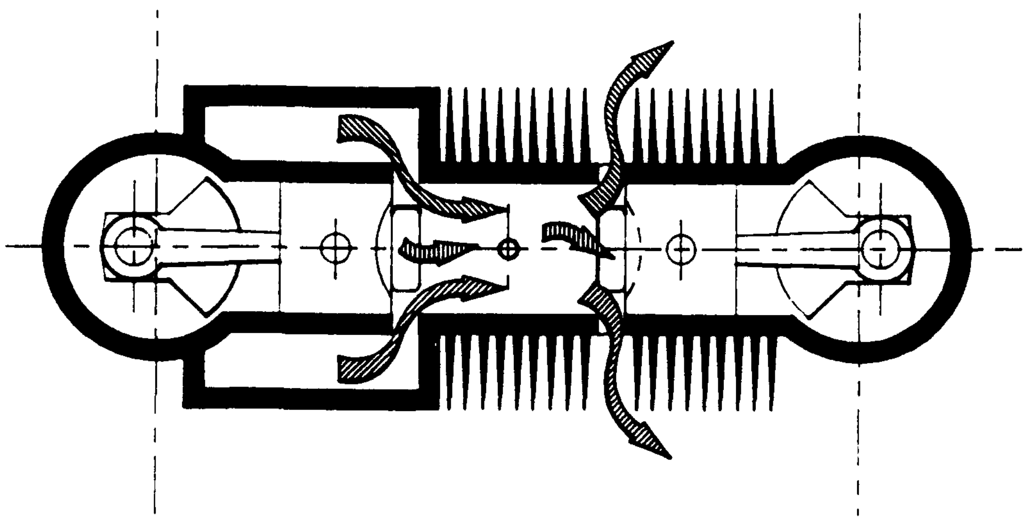


FIGURE 2. OPPOSED PISTON CONFIGURATION

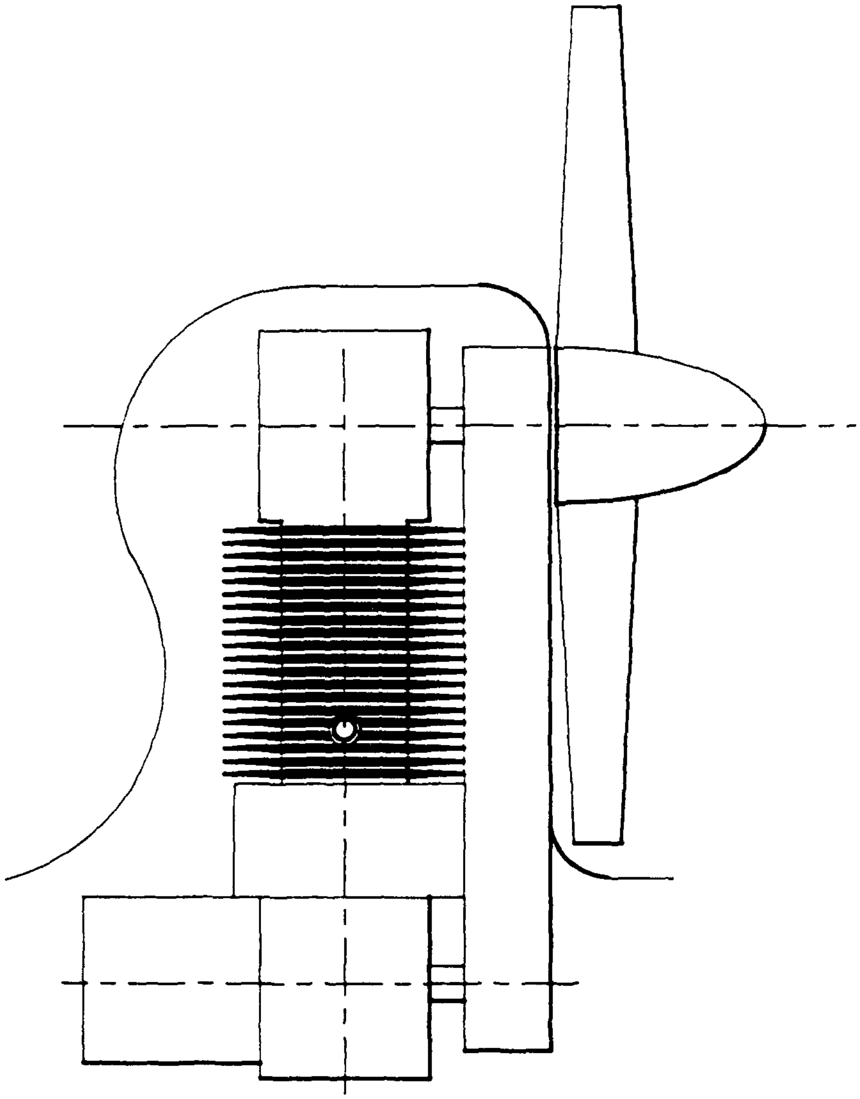


FIGURE 3. PYLON MOUNTING

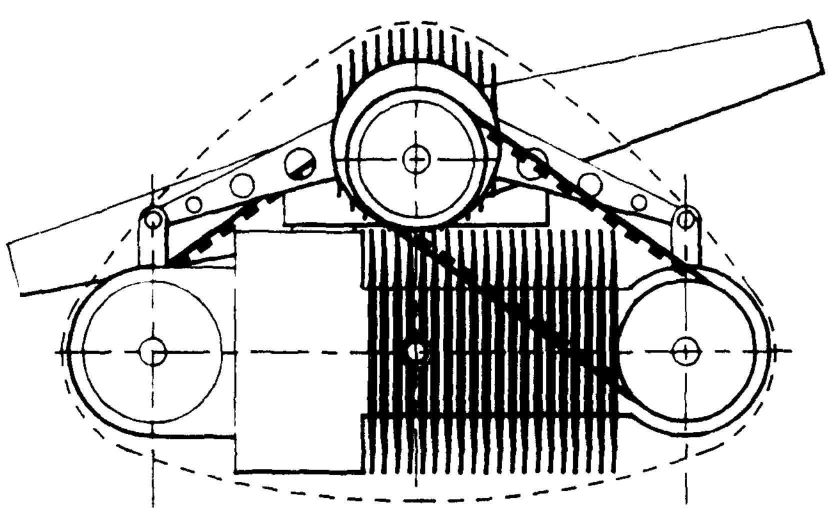


FIGURE 4. FUSELAGE MOUNTING

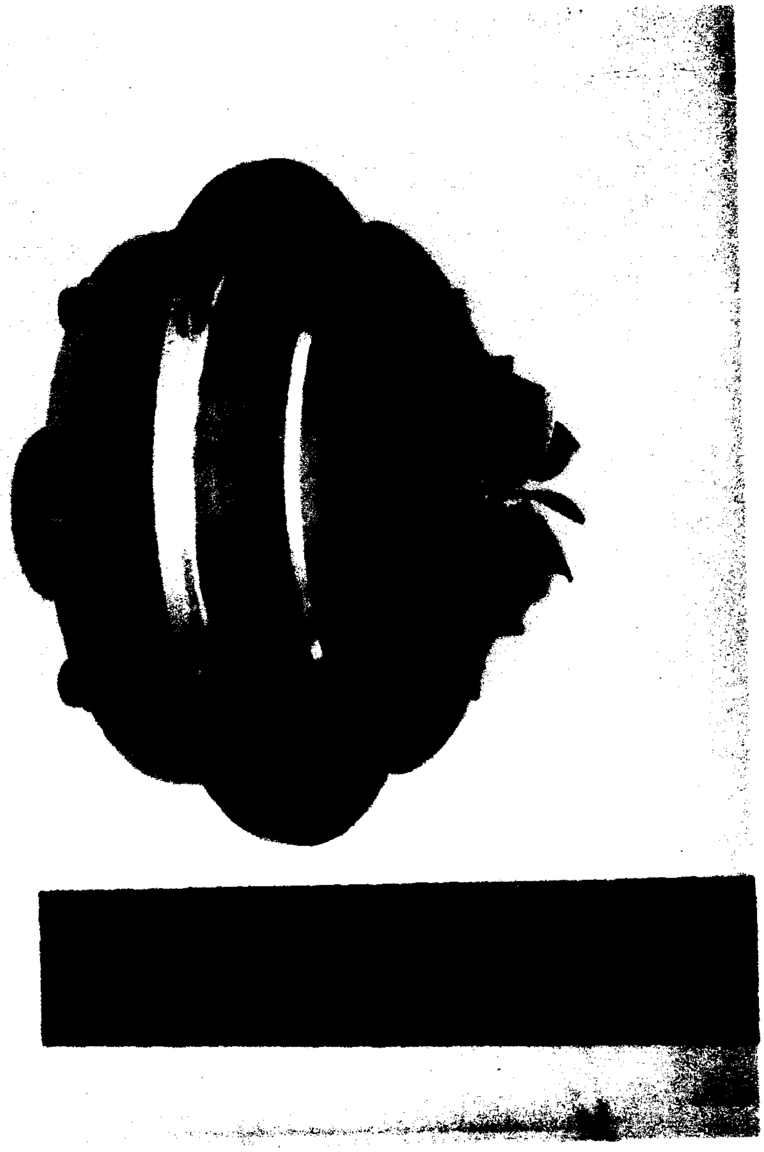


FIGURE 5. FRICTION DRIVEN CENTRIFUGAL COMPRESSOR

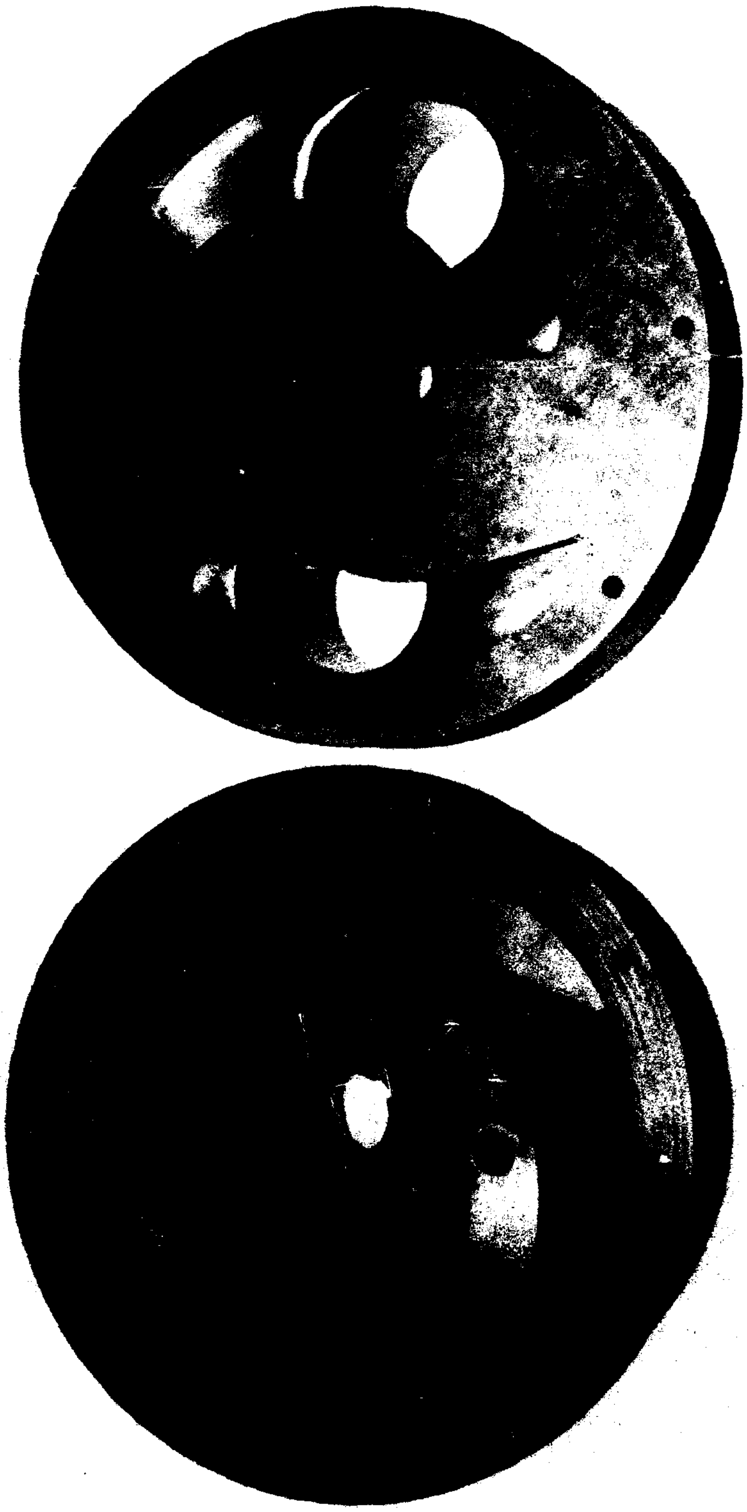


FIGURE 6. TRAILING VANE COMPRESSOR

FIGURE 7. SLIDING VANE COMPRESSOR

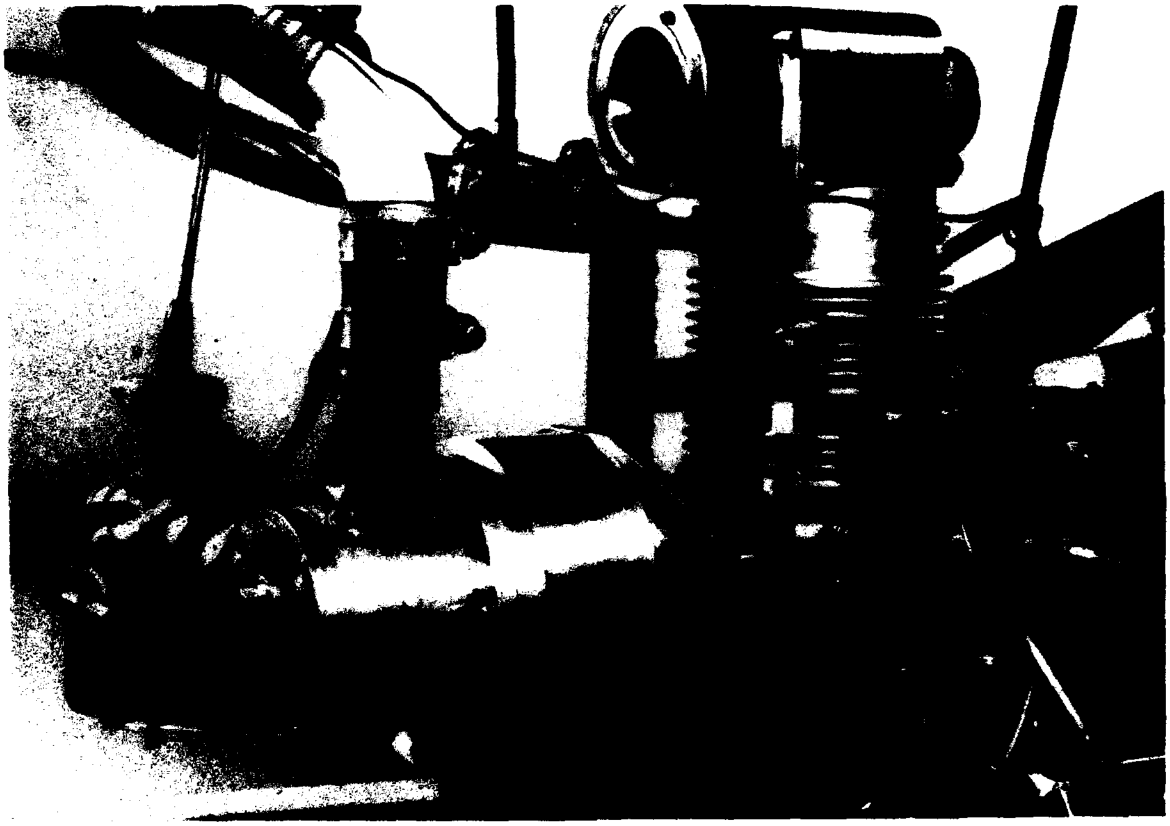


FIGURE 8. 67mL ENGINE ON TEST BED

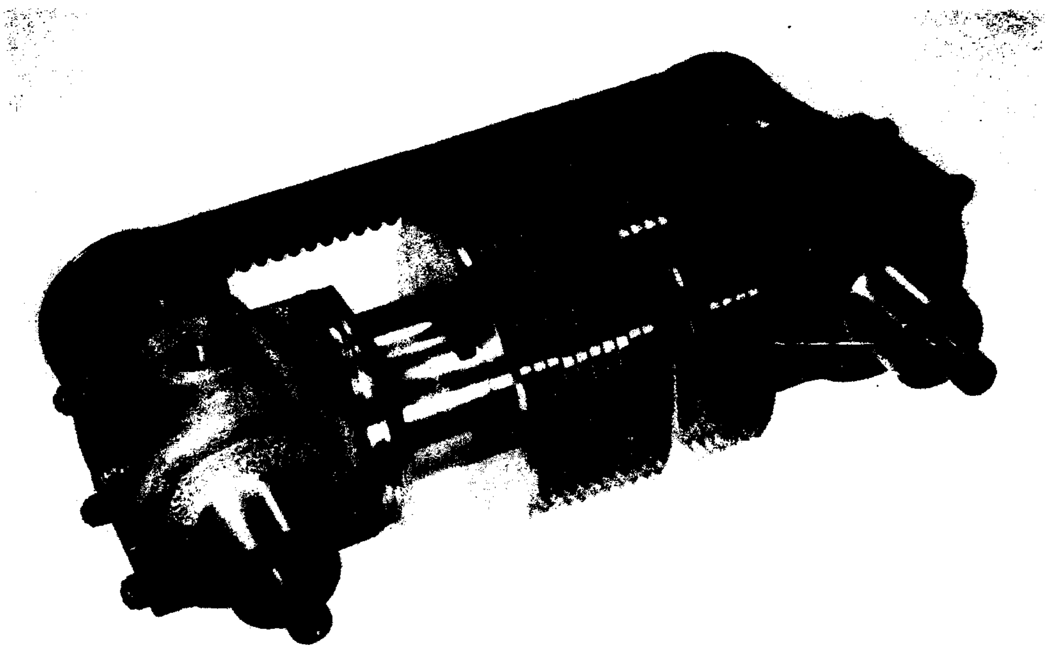


FIGURE 9. 225mL OPPOSED PISTON ENGINE

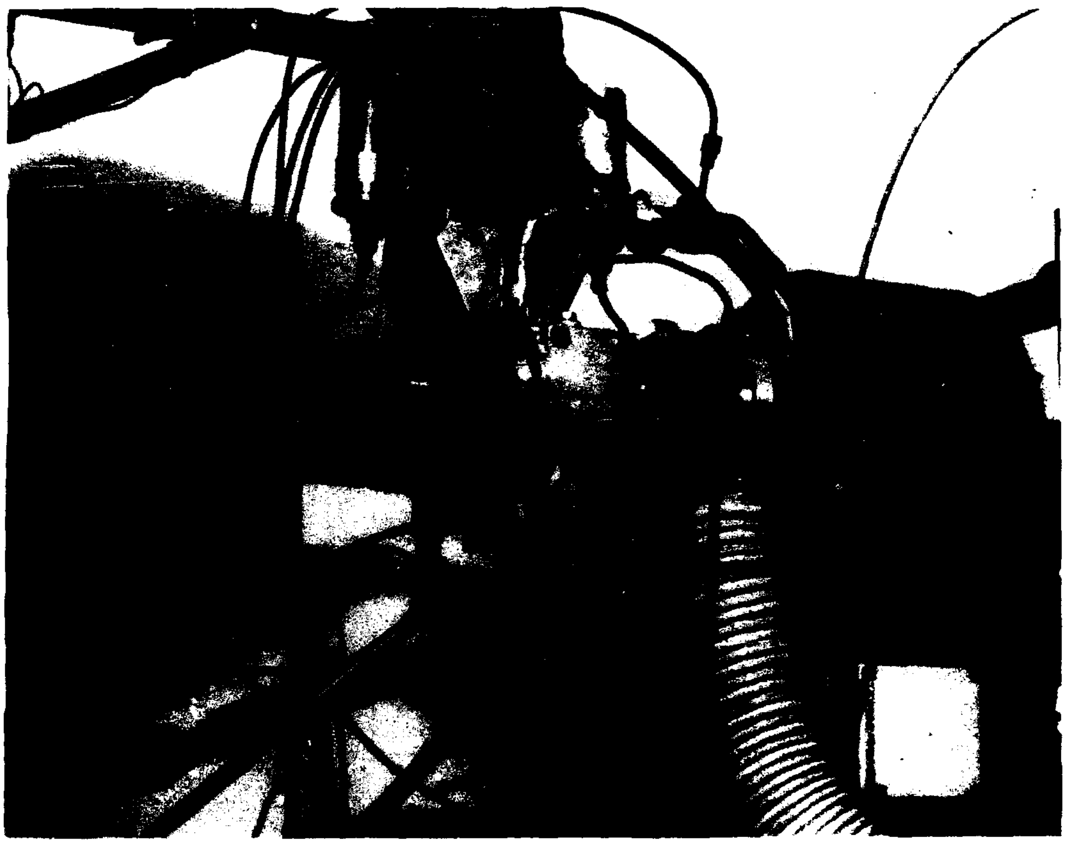


FIGURE 10. 225mL ENGINE ON TEST BED

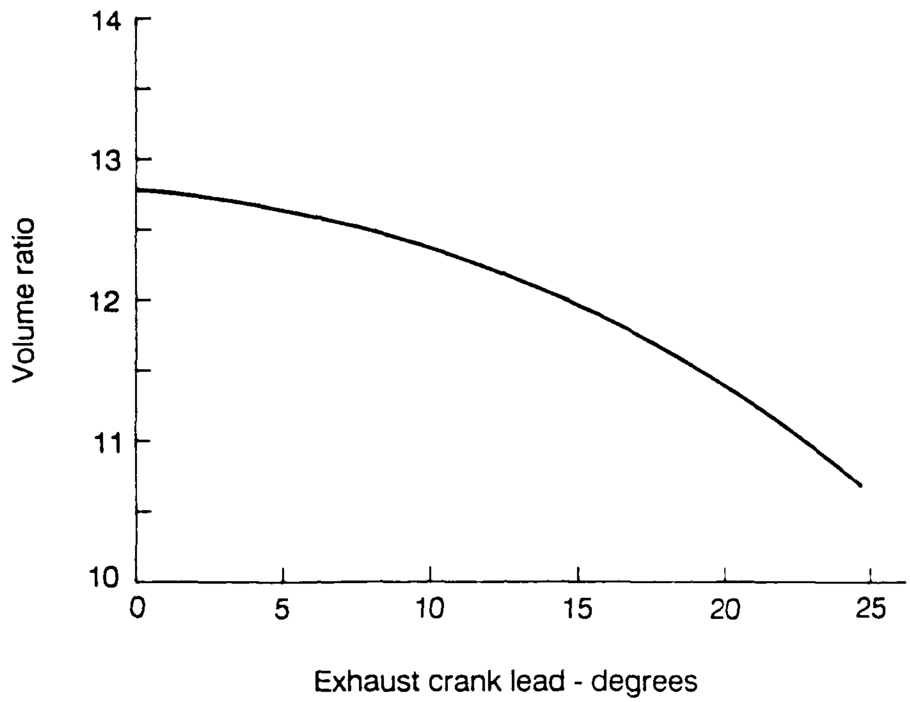


FIGURE 11(a). EFFECT OF CRANK PHASE ANGLE ON VOLUME COMPRESSION RATIO

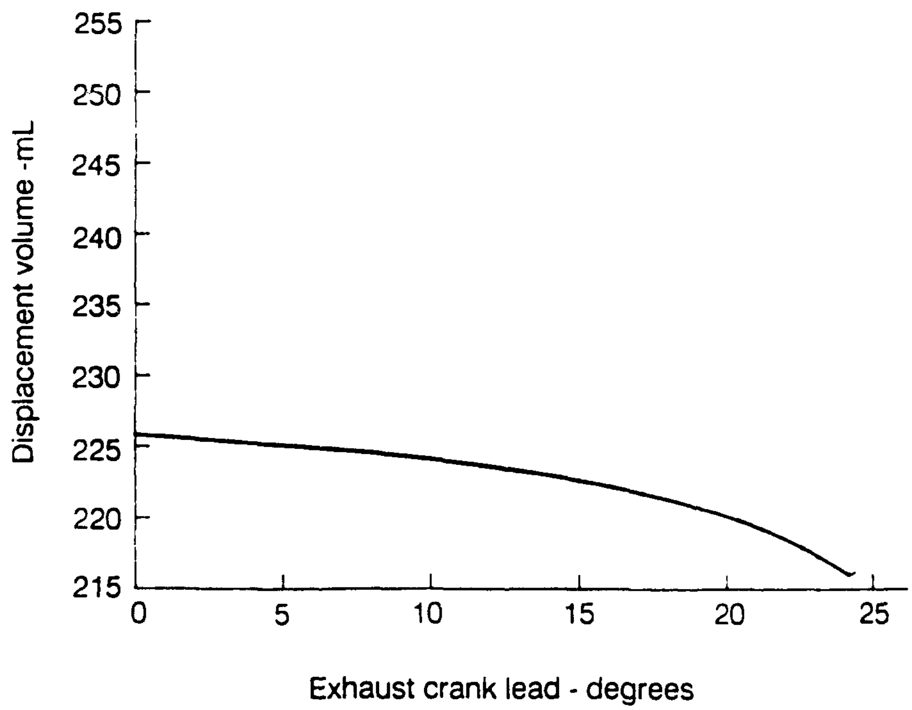


FIGURE 11(b). EFFECT OF CRANK PHASE ANGLE ON DISPLACEMENT VOLUME

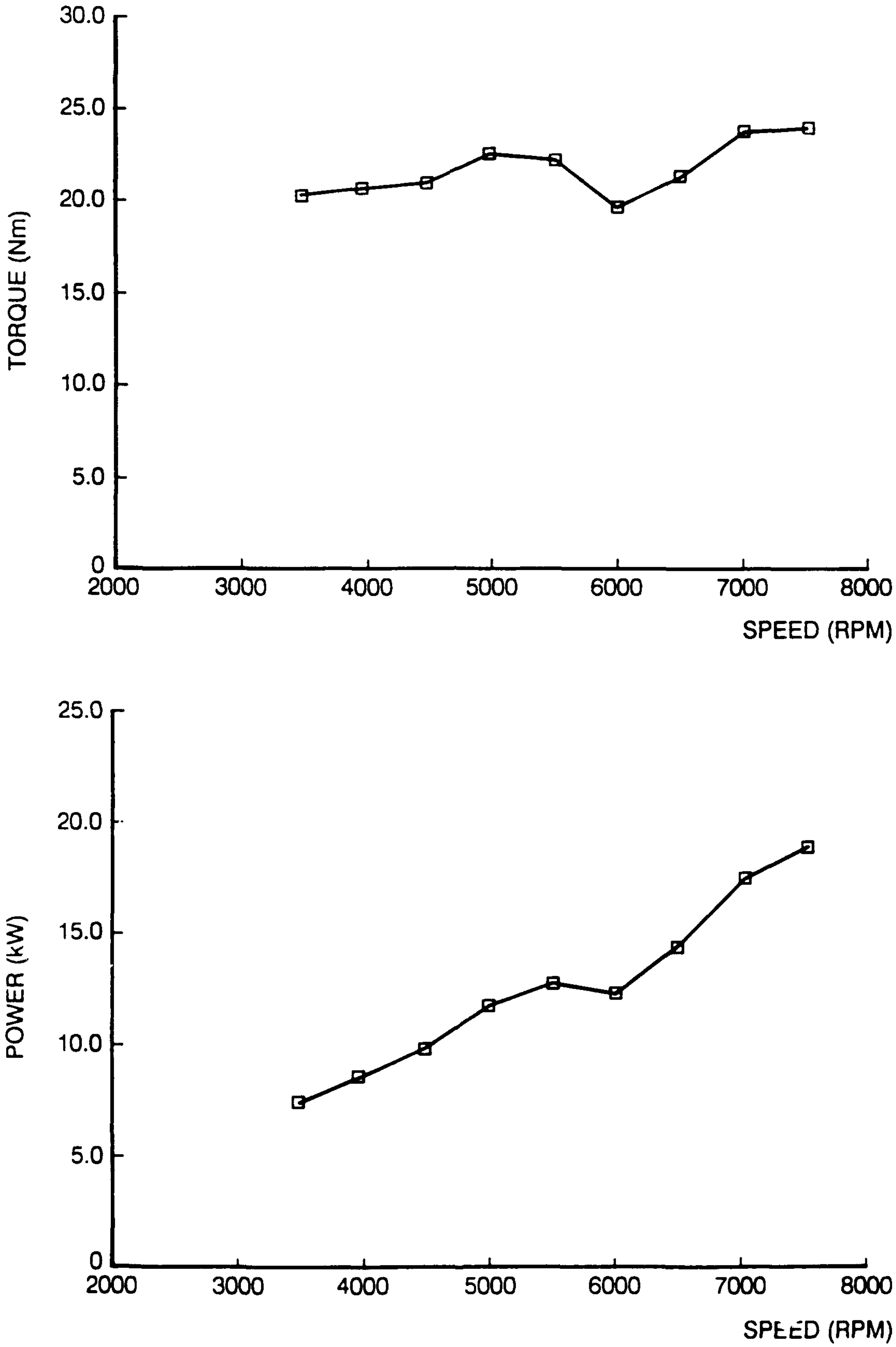
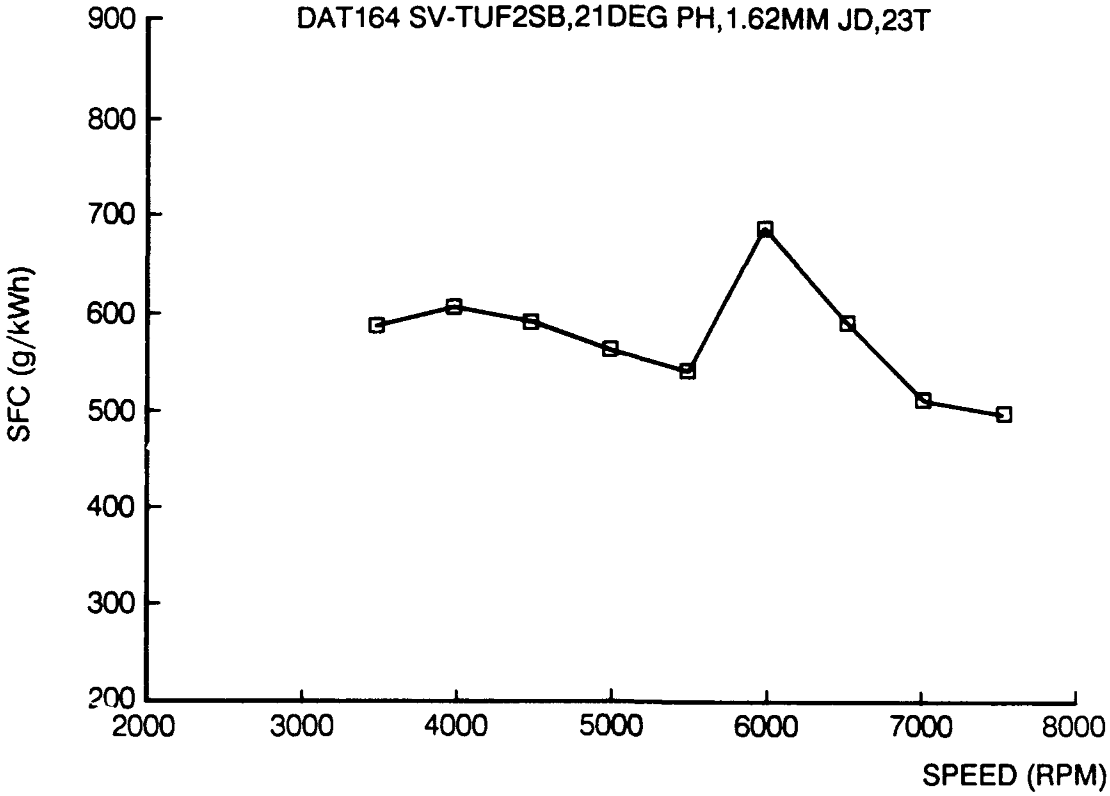


FIGURE 12 PERFORMANCE OF 225mL ENGINE



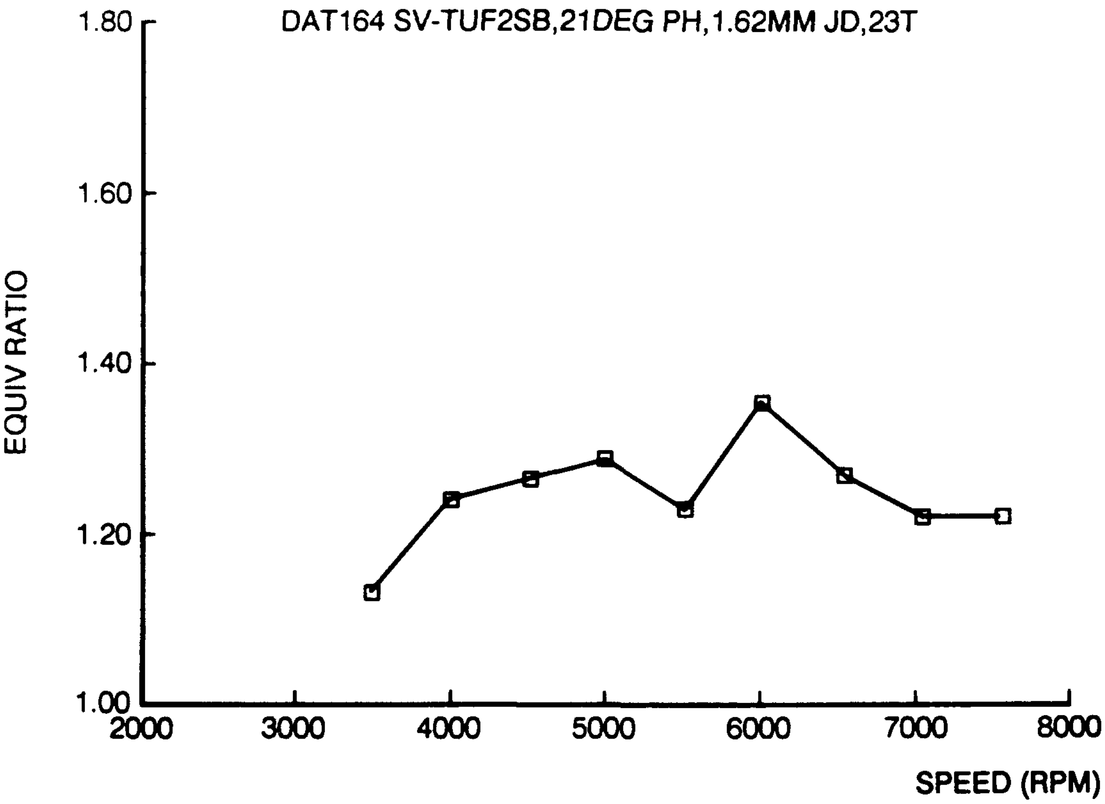
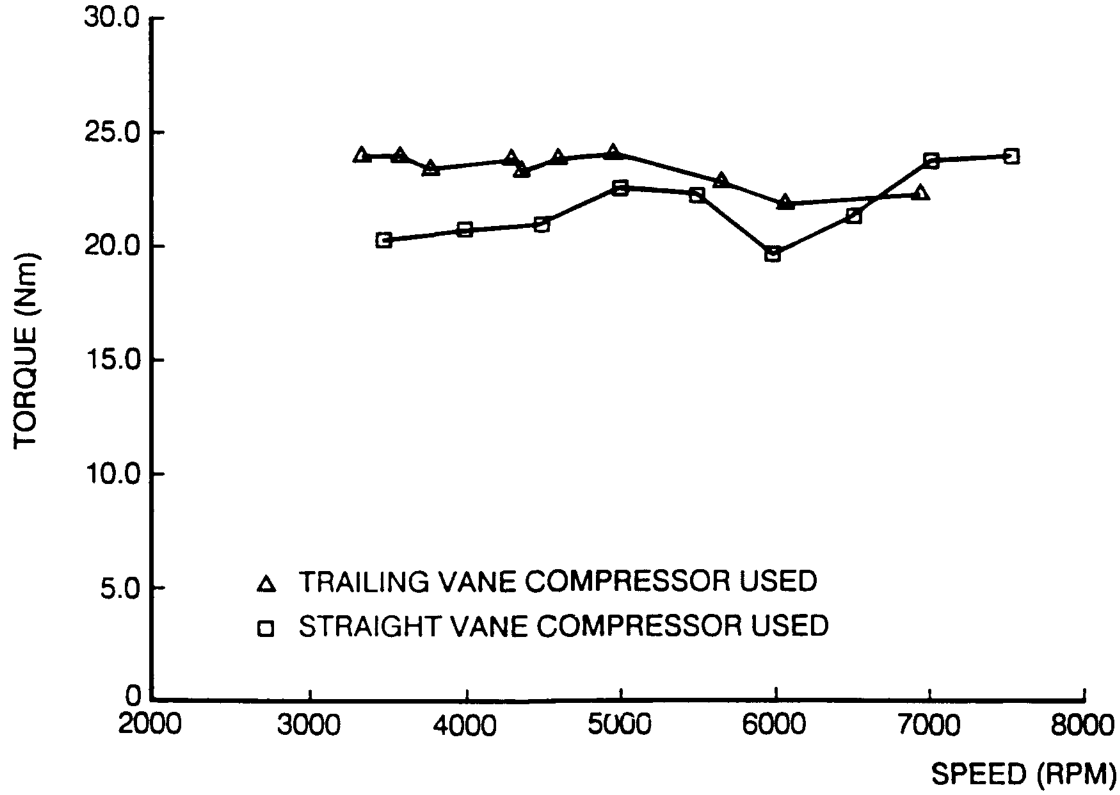


FIGURE 13 PERFORMANCE OF 225mL ENGINE (CONT.)



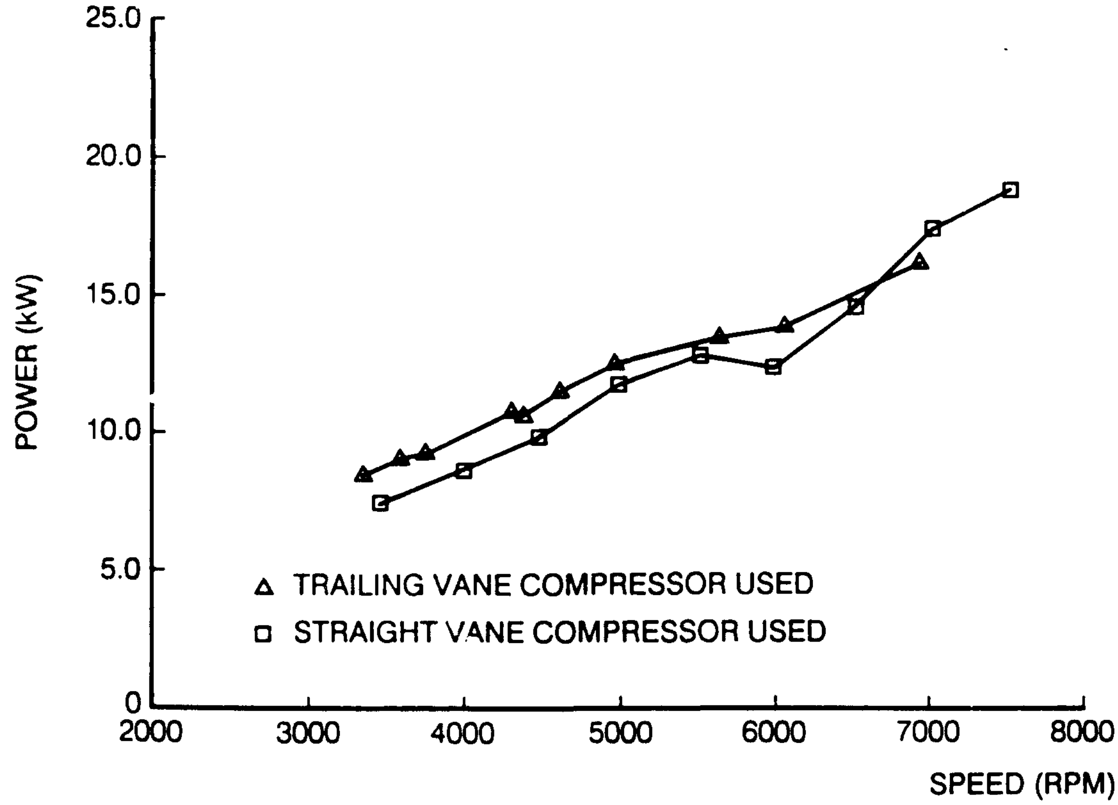


FIGURE 14 225mL ENGINE PERFORMANCE USING DIFFERENT COMPRESSORS

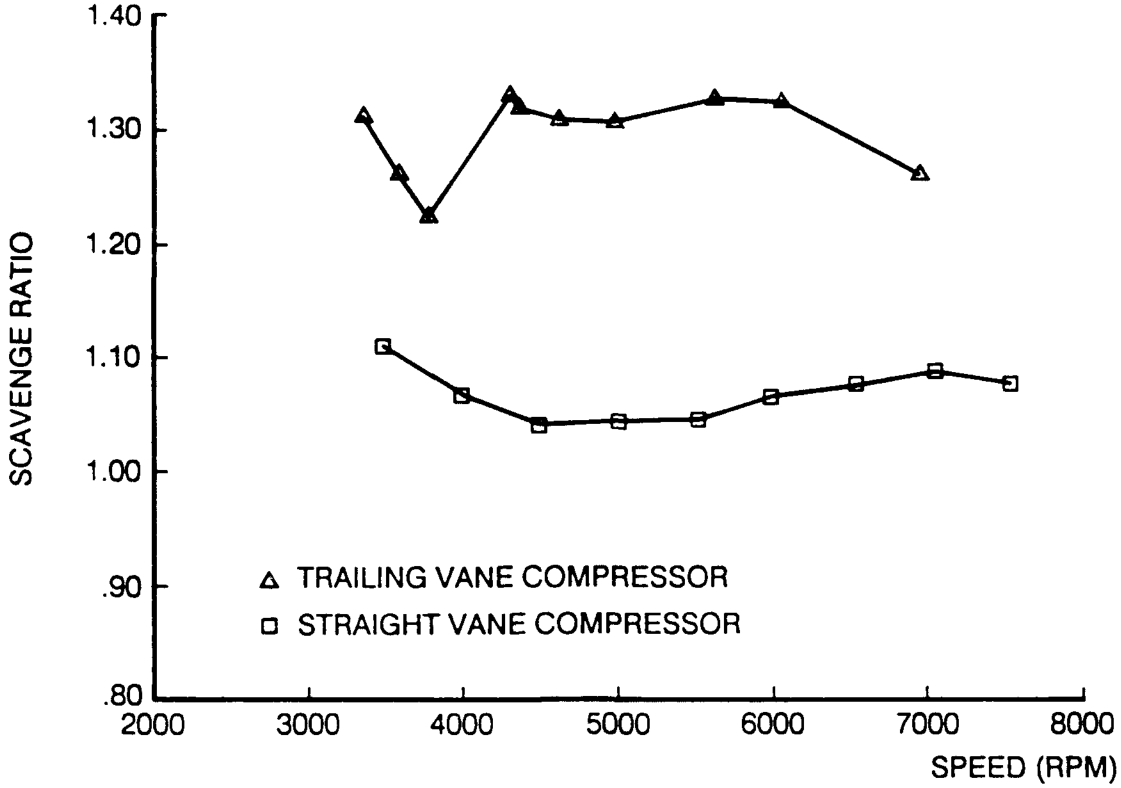


FIGURE 15 COMPRESSOR CAPACITY COMPARISON

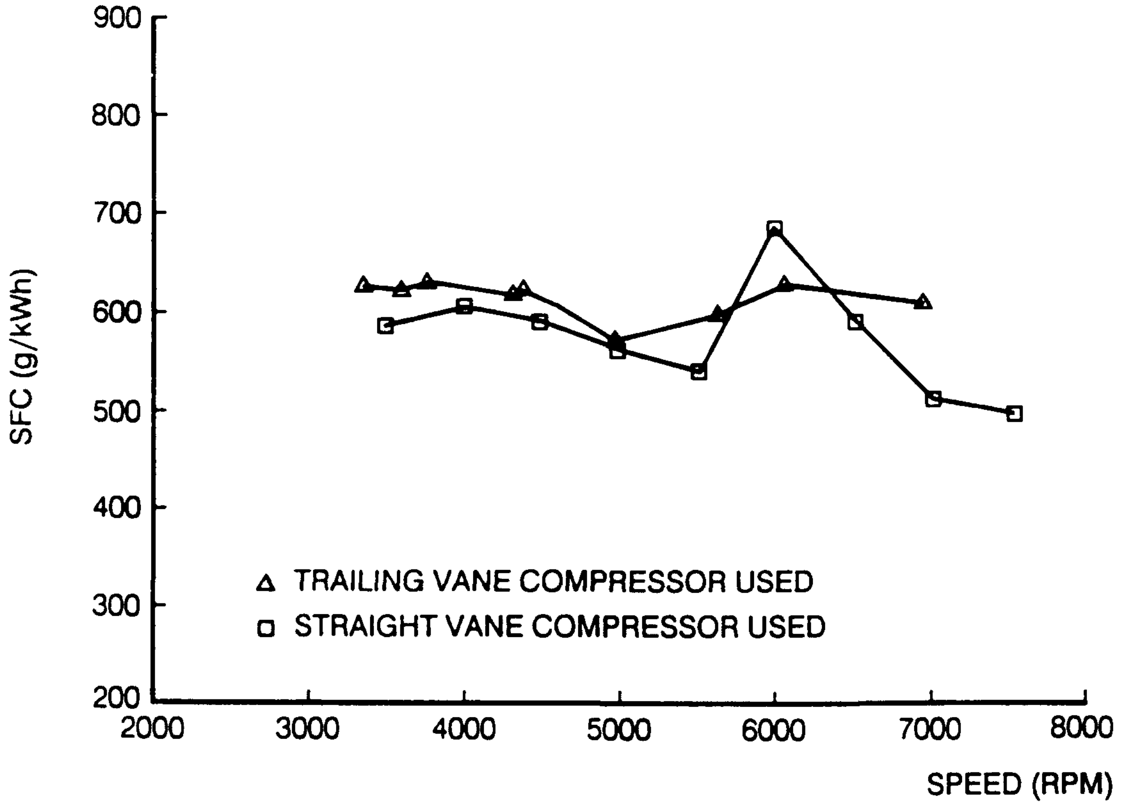


FIGURE 16 ENGINE PERFORMANCE USING DIFFERENT COMPRESSORS

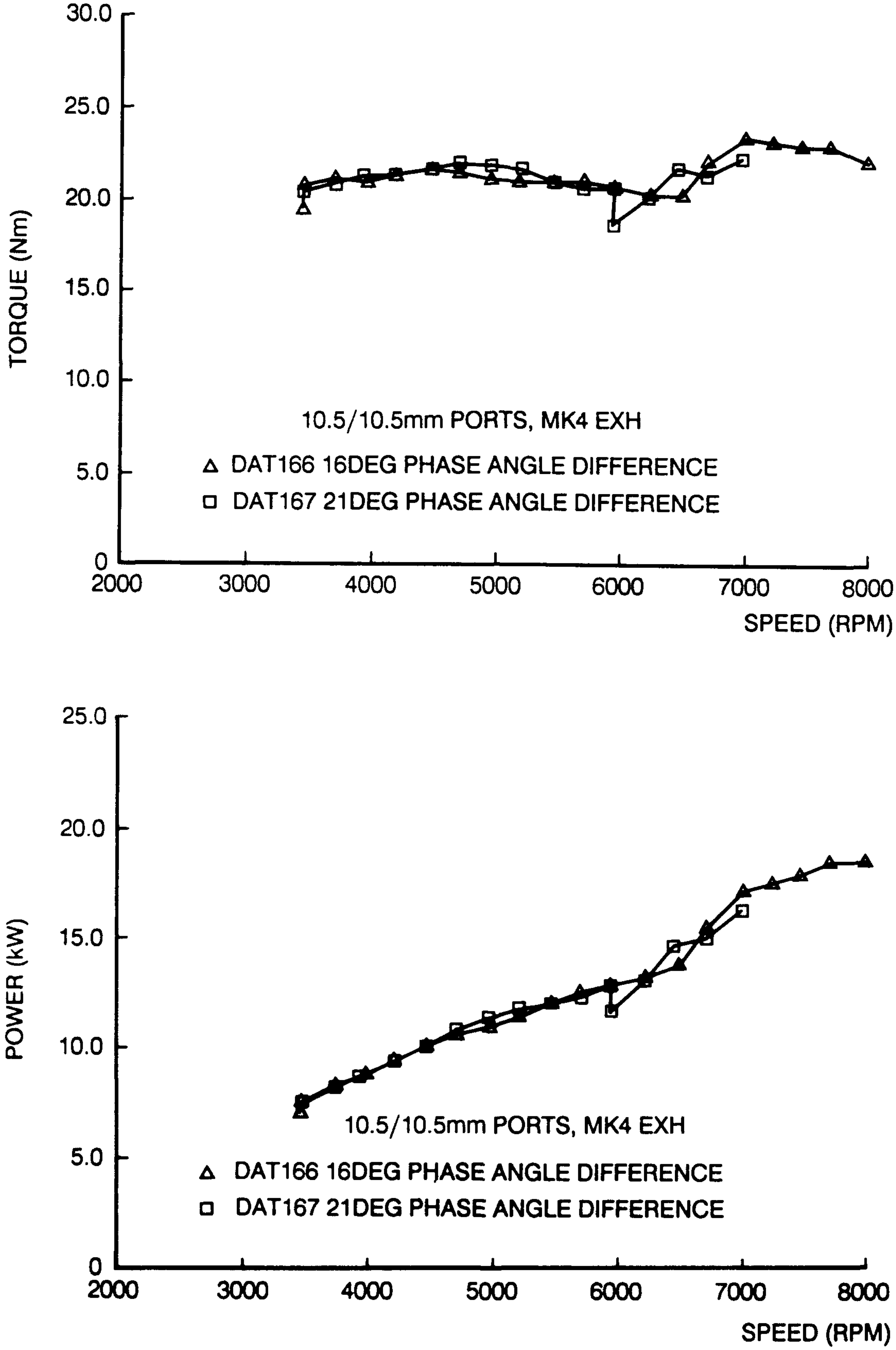
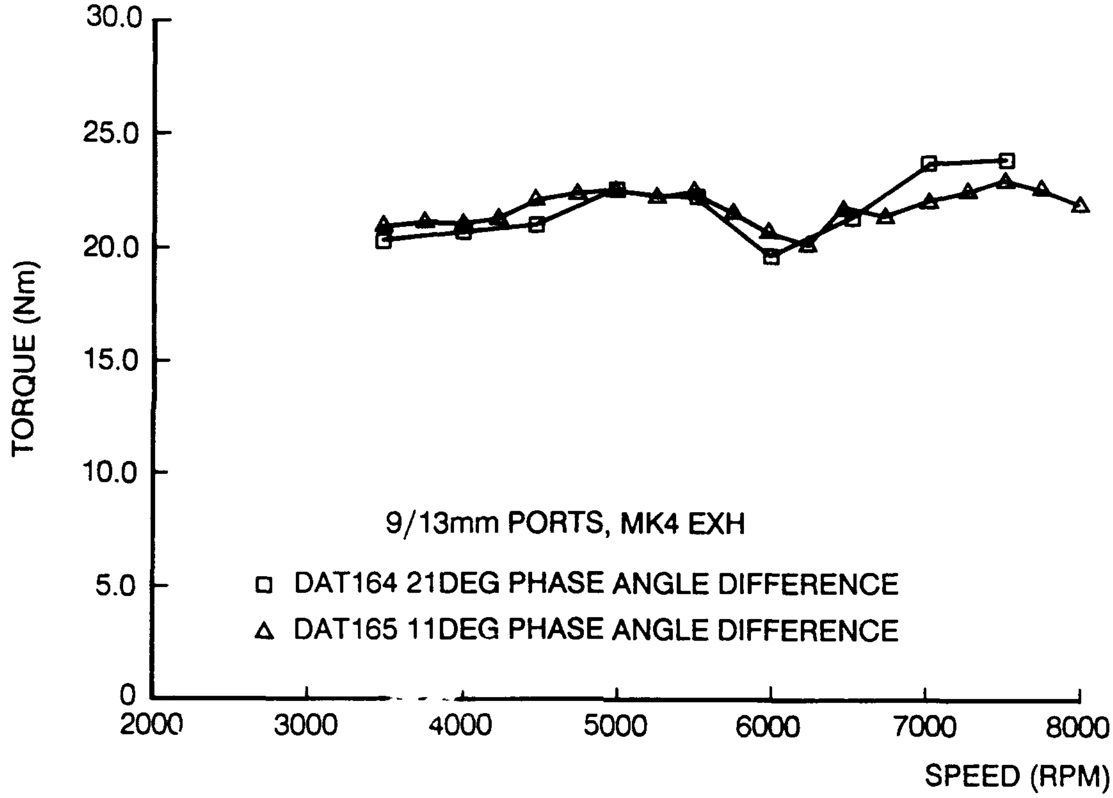


FIGURE 17 EFFECT OF PHASE ANGLE ON PERFORMANCE



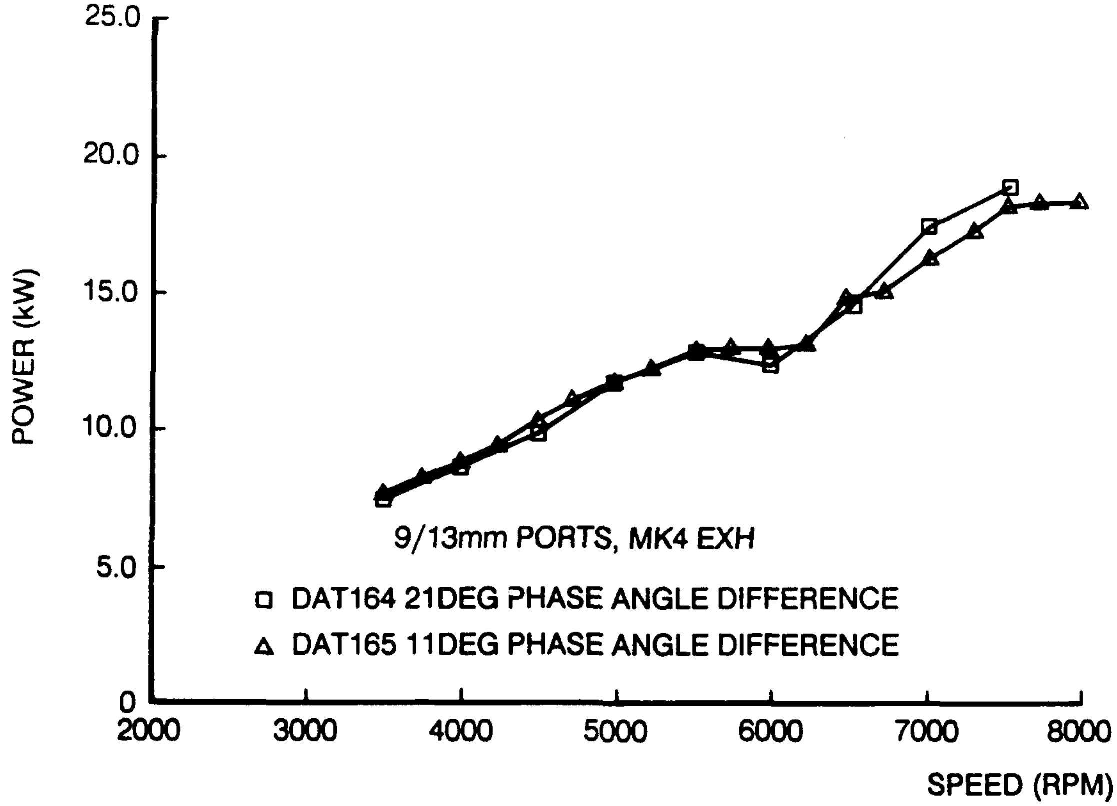
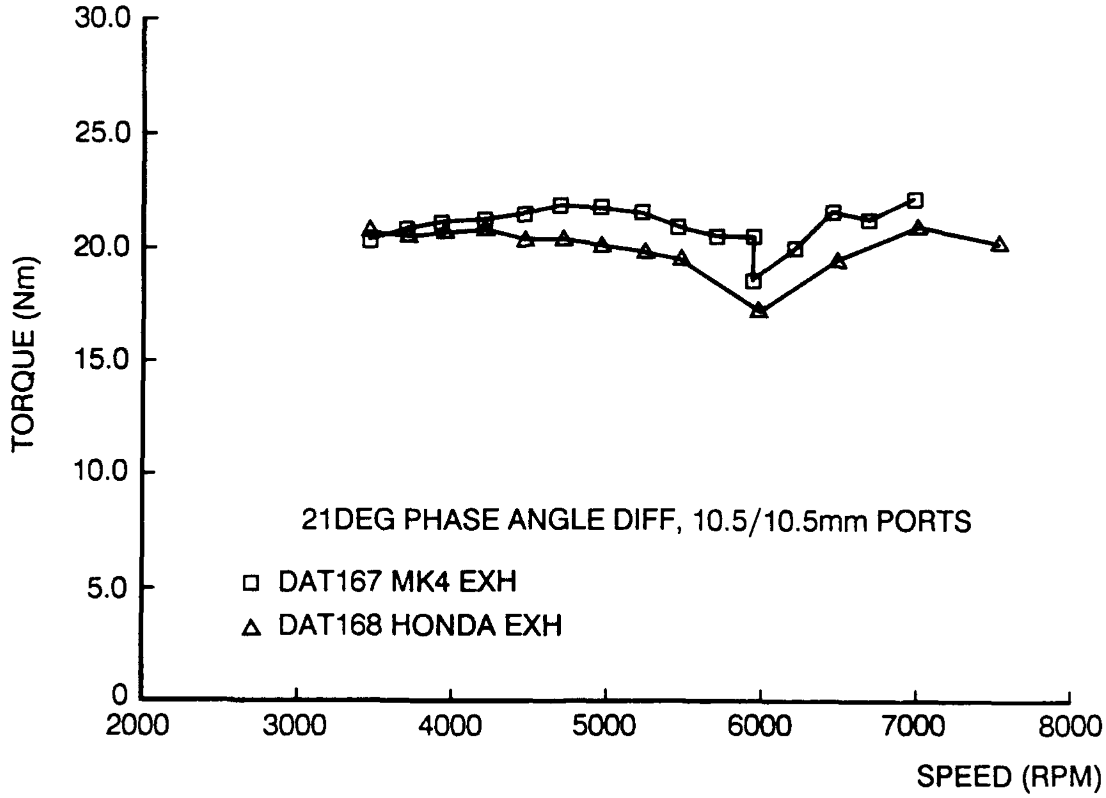


FIGURE 18 EFFECT OF PHASE ANGLE ON PERFORMANCE



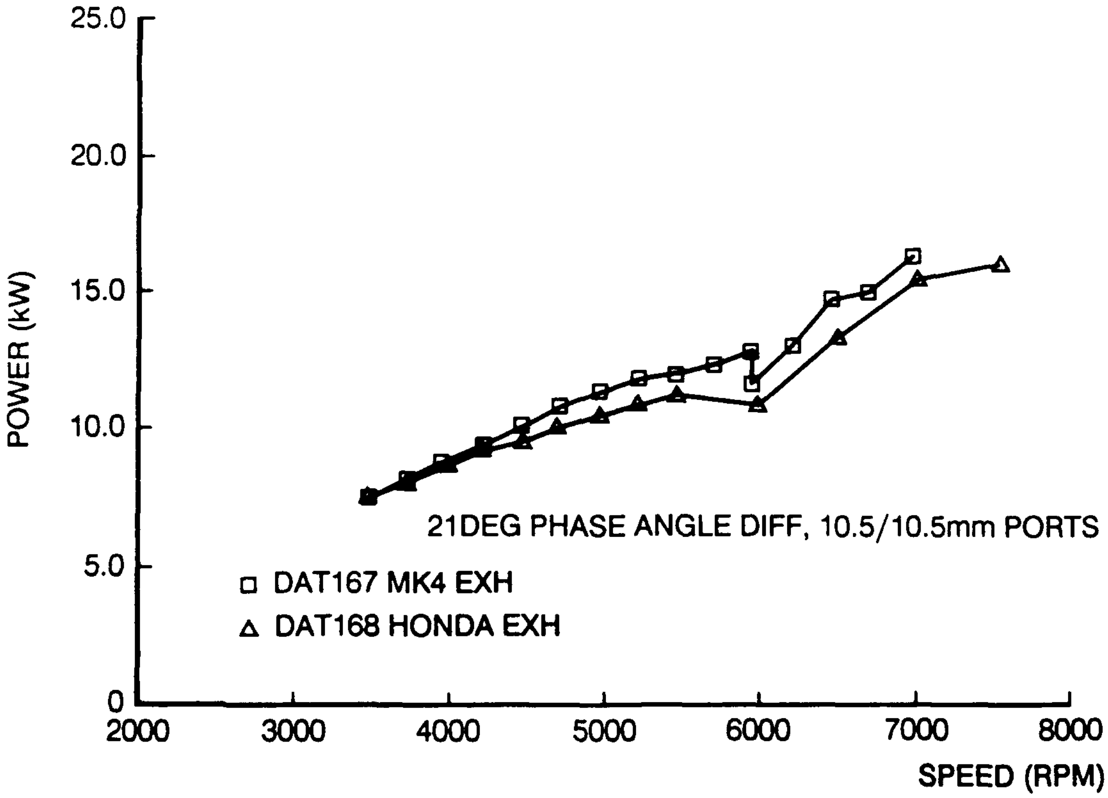


FIGURE 19 EFFECT OF EXHAUST CONFIG ON PERFORMANCE

APPENDIX 1

1. THE CAM ENGINE

Where small numbers of specially designed engines are required, as in the present case, one of the main difficulties facing the designer is the question of the crankshaft. It is most unlikely that any commercially produced crankshaft will be suitable for the purpose and the cost of manufacturing small numbers of cast, or more particularly, forged crankshafts is likely to be prohibitive. The usual solution to this problem is to use built-up crankshafts, assembled by shrink fitting the component parts. This is an expensive and difficult process and greatly complicates disassembly for replacement of the connecting rod or big end bearing should this be required.

An alternative to this is to use a bolted construction, as was the case for the opposed piston engine in the foregoing report. While this eases the problem of disassembly, it is still an expensive method of construction. During the course of refining the design of the bolted draw-plugs (which performed very reliably), Parmington proposed that savings in size, weight and cost could be achieved by eliminating the crankshaft altogether, and using in its place a simple cam mechanism. This would permit the camshaft to be manufactured from solid bar and would eliminate the connecting rod altogether, the load from the piston being applied to the shaft by means of a roller mounted on the gudgeon pin. The principle is shown diagrammatically in Figure 1-1.

The idea was prompted by the observation that, under normal operating conditions in two-stroke engines, the gas loads applied to the pistons exceed the inertia forces, that is, the connecting rods are under compressive load throughout the cycle.

It was recognised of course, that when the engine stopped, the pistons and followers would not remain in contact with the cams and that some arrangement to bring them into contact would be necessary when starting the engine. However, in the case of a UAV engine there is usually no requirement for restarting in flight so that, if the initial start could be accomplished reliably, advantage could be taken of the weight and cost savings expected to result from this design. The decision was therefore taken to construct a demonstration engine as part of the work described in the body of this report.

2. THE CAM MECHANISM

Selection of the cam profile is constrained by the requirement to maintain contact between the cam and the follower at all times in order to prevent impactive loading and bouncing. Since the maximum deceleration during the compression stroke will be determined by the available gas pressure, the choice of cam profile must take this into account. This consideration was

found to preclude the use of a constant acceleration profile, which would result in the lowest peak accelerations.

The cam profile finally selected for use on the demonstration engine was a cardioid, designed to impart simple harmonic motion to the piston. In this case the acceleration of the piston varies linearly with distance from the mid stroke position, whereas, on the compression stroke, the gas pressure rises exponentially, reaching a maximum value (in the absence of combustion) at the point of minimum volume.

Analysis shows that, neglecting friction, if the engine missfires, the critical part of the cycle occurs during the compression stroke, when the piston is about two thirds of the way up. At this point the reaction force between cam and follower falls to a minimum value and it is here that contact may be lost, possibly resulting in shock loads and bouncing of the follower on the cam. However, detailed calculations showed that, provided cardioid cams were used to give simple harmonic motion, and the mass of the reciprocating parts were minimised, contact between cam and follower could be maintained even in the event of misfiring.

Several alternatives were considered with a view to reducing costs by simplifying manufacture. These included the use of circular cams with circular followers and circular cams bearing on flat-bottomed pistons. Calculations indicated that the use of circular cams would result in high, but marginally acceptable values of piston acceleration. However, when used with flat-bottomed pistons, excessive tilting moments on the pistons as the point of contact moved away from the cylinder centreline would result.

3 PRELIMINARY TESTS

Because of the somewhat radical nature of the idea, the expense involved in a first assessment had to be minimised and the first test runs of the cam engine principle were carried out using components salvaged from the 67 mL crankshaft engine. Figure 1-2 shows the engine coupled to the water-brake dynamometer and Figure 1-3 shows the cam and follower. Pistons were turned from round bar and carried hardened steel cam followers mounted in needle roller bearings. For ease of manufacture the cams were circular although it was realised that this would result in higher piston accelerations than would be the case with cardioid followers.

In order to prevent the piston from rotating in the bore and to ensure that the cam and follower were kept in the same plane, a 3 mm diameter steel pin protruded from the cylinder wall and engaged a slot milled in the piston at 90 degrees to the thrust face.

Scavenge air was supplied by a very simple, early version of the trailing vane scavenge compressor. In all other respects the engine was as it had been in the crankshaft version.

The worn condition of the bore precluded any performance measurements. However, after setting the camshafts to the position corresponding to maximum volume between the pistons, and then blowing the pistons into contact with the cams by means of compressed air admitted through one of the spark plug holes, the engine could be started fairly readily.

During the start-up the followers could be heard rattling loudly on the cams, however, as soon as steady firing was established the engine ran smoothly although under very light load. It was started and stopped two or three times and accelerated quickly to 7000 rpm. At this speed a different rattling could be heard and the engine was stopped. Examination showed that the grooves used to keep the pistons correctly aligned were badly worn. Total running time had been of the order of about ten minutes.

Subsequent examination of the cams and followers showed them to be quite unmarked and this, together with the speed reached and the smoothness of operation was considered to warrant proceeding with the design of a cam version of the 225 mL engine, having the same bore, stroke, and port timing, with the objective of demonstrating that, by the use of the cam mechanism, the same performance could be obtained from an engine which was about 15% smaller and lighter.

4 225 mL CAM ENGINE

The general arrangement of the engine is shown in Figure 1-4. The smaller size compared with the crank engine may be seen in Figure 1-5. As mentioned above, cardioid cams were used and these were on camshafts whose centrelines were offset from the cylinder axis so as to reduce the side thrust on the piston during the firing stroke. (Later unpublished calculations by Brizuela, showed this to be unnecessary.)

In order to reduce the cost of construction and also to improve heat transfer from the cylinder, the need for a cylinder liner was eliminated by the use of a special Comalco aluminium alloy (designated 3HA) to make a single casting incorporating the cooling fins, bearing supports, the inlet plenum chamber and a mounting plate for the scavenge compressor. This alloy is specially formulated to allow the pistons to be run straight on the bore, without any surface treatment.

Rotation of the pistons was prevented by the use of two hollow steel rods mounted in each of the cam housings as shown and engaging in holes in bosses cast into the piston skirt. Small nozzles fitted to the end of these tubes enabled lubricating oil to be squirted on to the underside of the piston crowns in order to achieve some cooling effect. This was considered advisable in view of the fact that the more compact dimensions compared with the crank engine reduced the length of cylinder wall available for cooling fins.

A trailing vane air compressor supplied the scavenge air through a passage cast integrally with the mounting plate.

As with the crankshaft engine, the shafts were mounted in needle roller bearings and were interconnected by a toothed belt. The eddy current dynamometer was coupled to the exhaust crankshaft.

5 TEST RUNNING WITH 225 mL CAM ENGINE

The technique used for starting the engine was the same as that used with the smaller engine. Again the engine started fairly readily and ran quietly and smoothly, once steady firing was established.

After running for only two or three minutes it was dismantled and examined. No evidence of distress was to be seen on either the cams or followers. It was found necessary to slightly radius the ends of the piston aligning rods in order to prevent them scoring their grooves in the pistons.

In a subsequent short test the engine was accelerated cautiously to about 4200 rpm under no load. At this speed it stopped abruptly and examination showed that, in both pistons, the bosses which retained the cam follower bearings had failed. This caused the cam followers to become misaligned with the cams so that they jammed and stopped the engine.

This failure probably resulted from an error in the design of the pistons, in that the load was carried in bending in the side walls rather than being transferred to the crown by stiffening webs. (The piston was designed this way in an attempt to limit the heat transfer from the crown into the cam follower bearings.) The failure should not be seen as an inevitable result of the basic concept.

Although further design studies were carried out on this concept, higher priorities accorded the conventional opposed piston engine prevented further test bed running.

6 CONCLUSION

Clearly, some further development is required before the principle could be used to build a useful engine. However, in the opinion of the authors, the basic feasibility of a reciprocating two-stroke engine in which the pistons are returned to the bottom of the stroke without the use of a mechanical linkage, that is, by the gas forces alone, has been demonstrated.

As in the case of the crankshaft engine, the use of direct fuel injection may well permit the use of symmetrical port timing, in which case the engine could be completely dynamically balanced.

The most likely area of application is to a small expendable two-stroke engine for military use, where the low level of vibration and the reduction in size and weight made possible by the use of the cam mechanism could well be of critical importance.

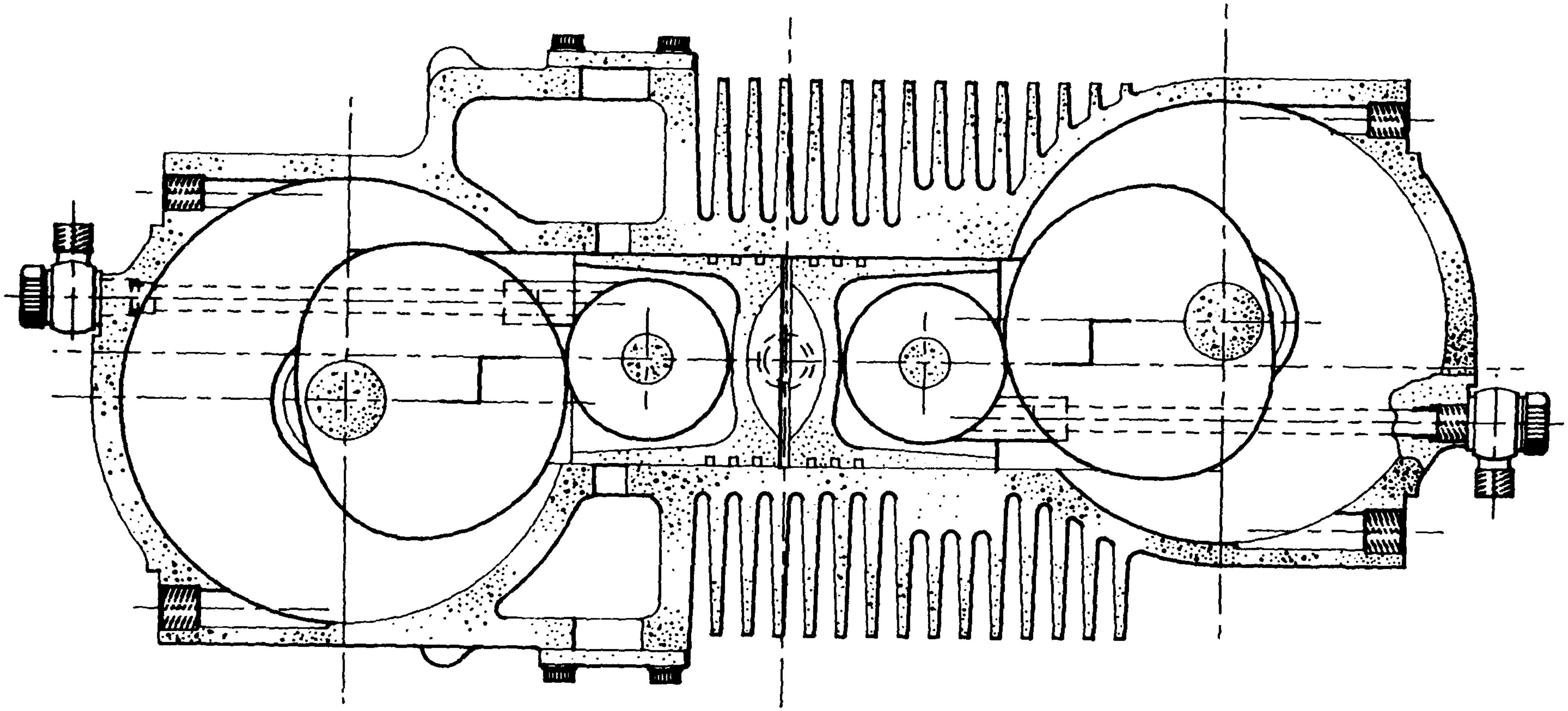


FIGURE 1-1 OPPOSED PISTON CAM ENGINE

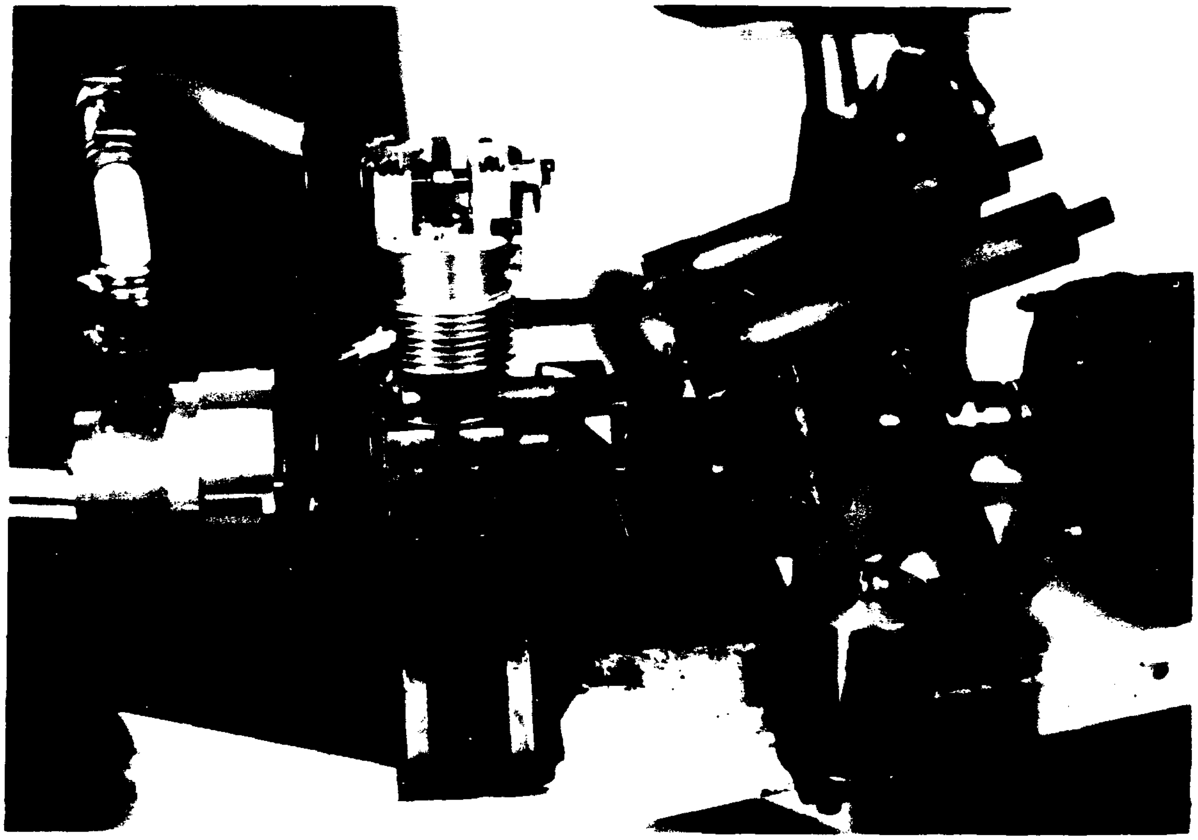


FIGURE 1-2 67mL CAM ENGINE COUPLED TO WATER BRAKE DYNAMOMETER

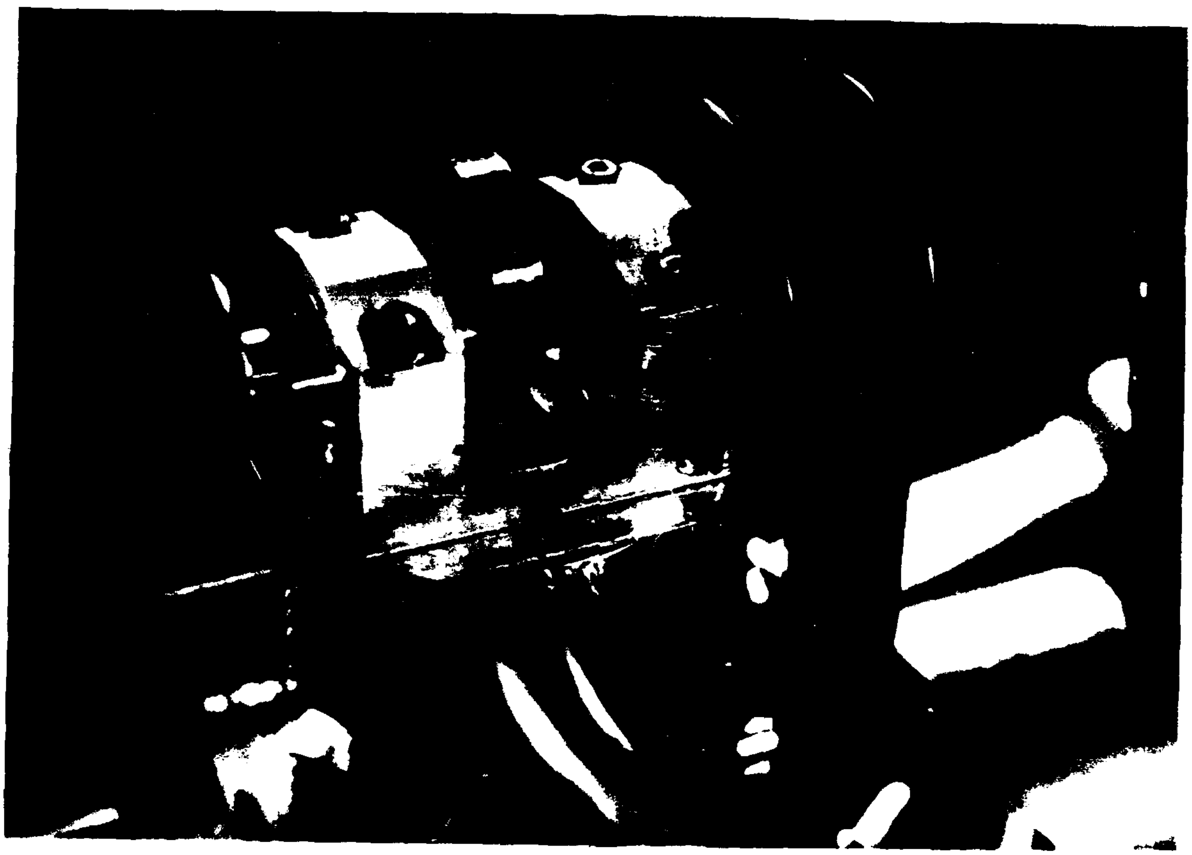


FIGURE 1-3 67mL CAM ENGINE - CAM AND FOLLOWER

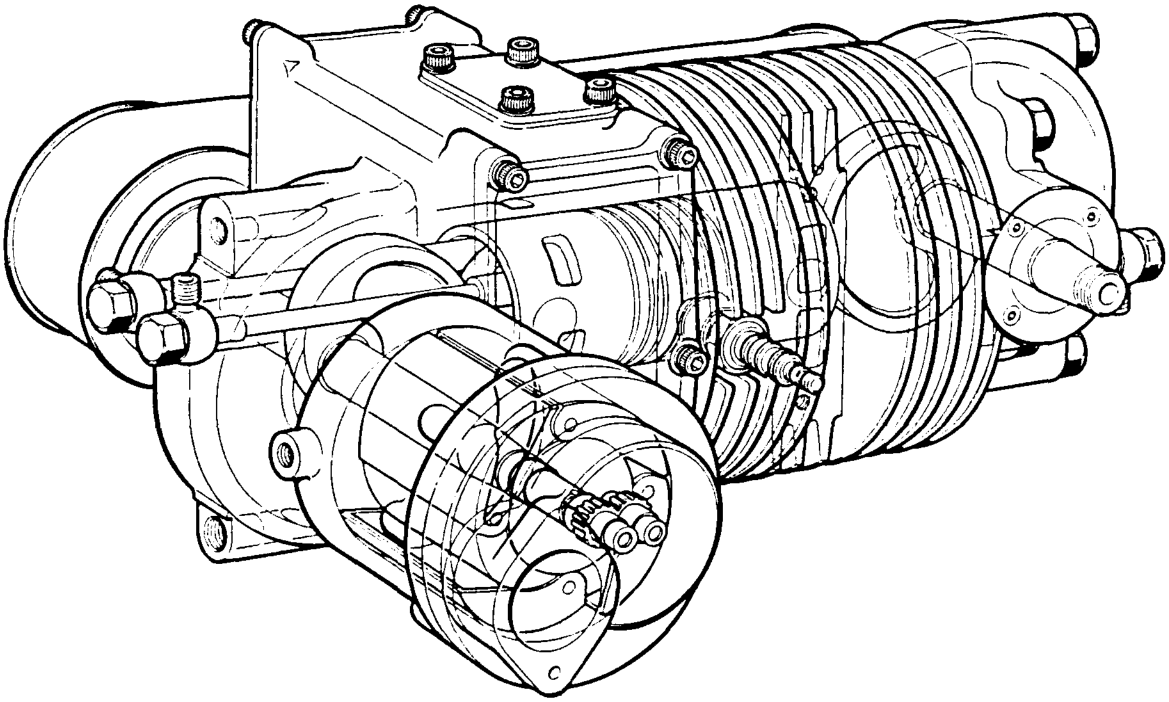


FIGURE 1-4 GHOST VIEW OF 225mL CAM ENGINE

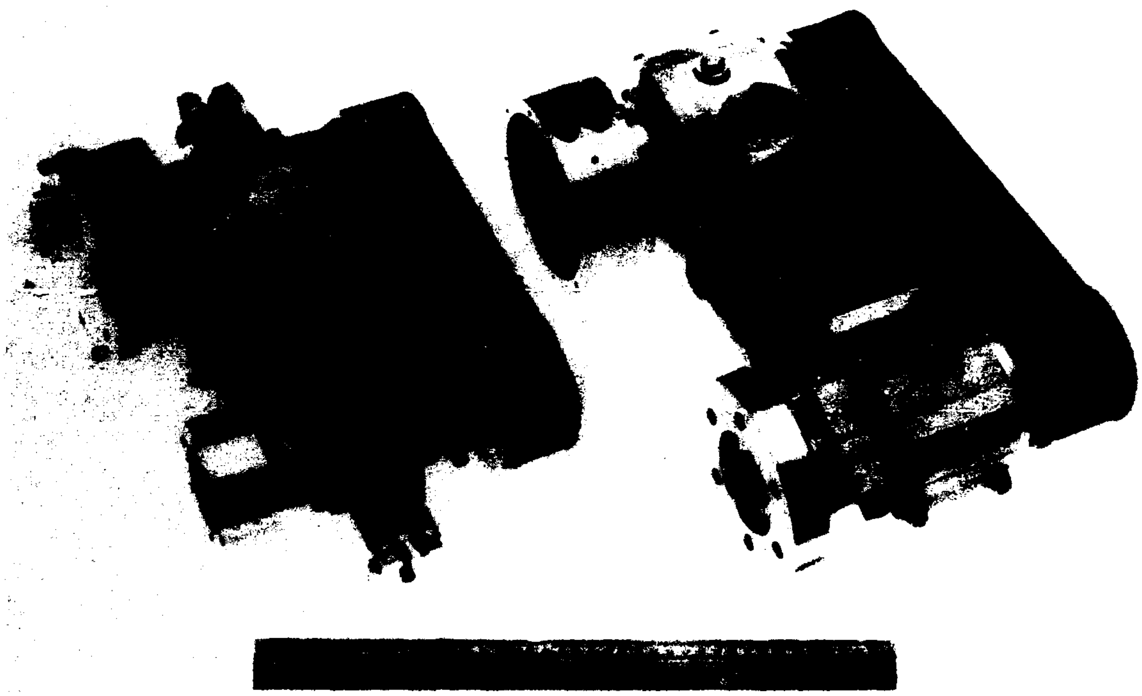


FIGURE 1-5 225mL CAM ENGINE (LEFT) COMPARED WITH 225mL CRANK ENGINE

DISTRIBUTION

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This report describes a program of work aimed at demonstrating the advantages of the opposed piston two stroke engine for use in small, propeller-driven, unmanned air vehicles requiring power of about 10 to 20 kW. Analysis of the requirements for engines for this duty showed the opposed piston configuration as offering the best compromise in terms of specific output, specific fuel consumption, level of vibration and complexity.

Initial feasibility was demonstrated using a small engine of 67mL swept volume. The major part of the development, including selection and development of a scavenge air compressor, was carried out using an engine of 225 mL swept volume. The maximum power obtained was 18.9 kW (25 HP) at 7500 rpm and the engine is considered to have potential for considerable further development.

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16. ABSTRACT (CONT).

Considerations involved in the selection of the configuration are discussed. The development of the rig engine and of the scavenge air blower is described and details of the performance are given. Factors limiting the performance and the potential for increased output and improved economy are outlined.

During the course of this work the initial feasibility of a more radical design, with the connecting rods and crankshafts replaced by a system of cams and rollers, was also demonstrated by making and running an engine. This arrangement offers the prospect of saving about 15% in size and weight compared to the conventional engine of the same swept volume.

Work on the cam engine is described in an appendix to the main report.

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